

Vol. 3, No. 2 BENTLY NEVADA

ORBIT

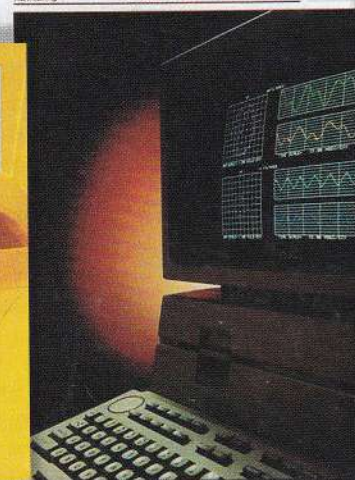


Critical Machinery

Today's economic environment demands monitoring general purpose as well as critical machinery.

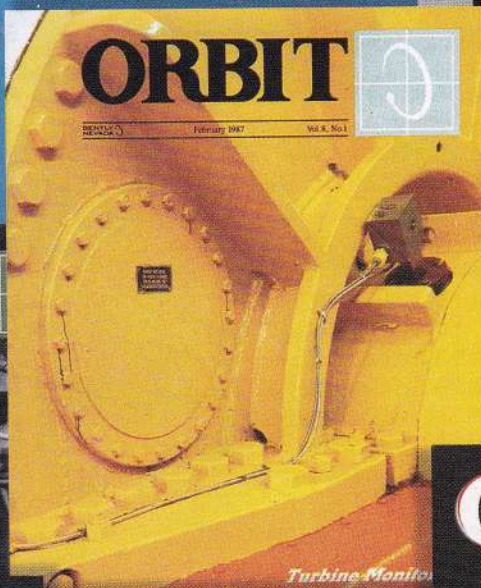
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December 1984 Vol. 5, No. 3



ORBIT

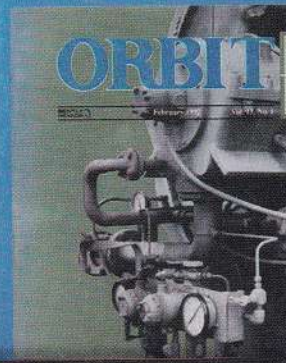
February 1987 Vol. 8, No. 1



Turbine Monitor

ORBIT

February 1987 Vol. 8, No. 1

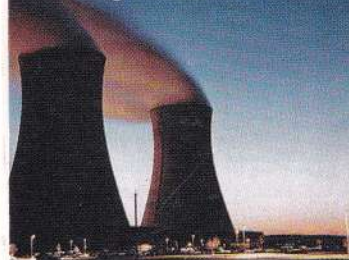


ORBIT

September 1997 Vol. 18, No. 4

ing a nuclear plant's
ing performance
and efficiency

PECO Energy is committed
onal excellence is through
machinery management



ORBIT

April 1985 Vol. 6, No. 1



Bently Nevada's
Team of
Service Experts

ORBIT

SECOND/THIRD QUARTERS 1999 VOL. 20 NO. 2

FEATURE: TRENCH PROBLEMS

How Product: An actively
recoiled ball bearing

Magnetic bearings - do
they problems outweigh
their benefits?

Cost History: Using Data
History: 2000 to compare
to the wholesale power
market

Why phase information is
important for diagnosing
machinery problems

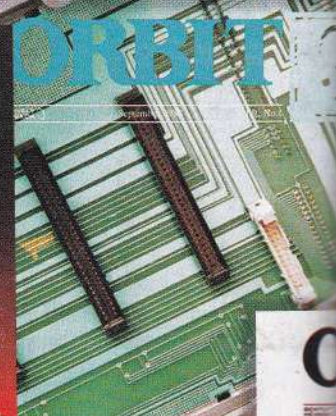
Update: Sagemat for
Windows CE - Bently
Nevada's next generation
portable data collector

Getting to know Bently
Nevada's management
team - Part 4

...all
your machinery

- portable systems
- on-line scanning systems
- fluid-film bearings
- active bearings
- machinery management
- on-line systems
- proximity probes
- permanent systems
- training
- vibration
- services
- off-line systems
- portable data collection
- proximity probes
- essential machinery
- temperatures
- flows
- position
- overspeed
- expansion
- casting transducers
- machinery protection
- general-purpose machinery
- critical machinery
- systems integration
- continuous systems
- rotary element bearings

HELPING YOU PROTECT AND MANAGE YOUR MACHINERY



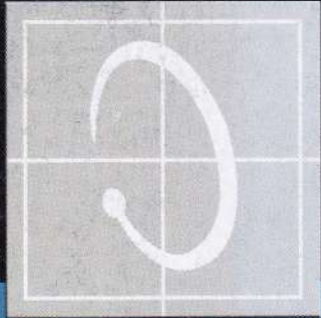
ORBIT

March 1983 Vol. 4, No. 1



ORBIT

FIRST QUARTER 2000 VOL. 21 NO. 1



the Orbit

Published by Bently Nevada Corporation

Winter 1981

CUSTOM PRODUCTS: MEETING CUSTOMER NEEDS

Its founding in 1961, Bently has proved itself as an ability to its customers with the special products they needed. If it wasn't in the catalog, as a standard item, they could be modified to meet it. It could be something new, something old, something in between. A few years later, the different demand was becoming distinct of the growing volume of requests.

Nevada responded to this demand by creating the Custom Products Dept. — a company within a company — to take over the manufacture of "OD" or "over the door" custom products. Nevada's Custom Products Dept. handles four divisions: Service, Planning, Engineering, and Manufacturing, and Quality.



Custom Products of Nevada Corporation

me, and it is staffed with a mix of design engineers, plan engineers, production and support. Jim Prevette, manager, says

"It's not available, then we build it ourselves." A walk through the facility reflects the enthusiasm of the CP manager. Technicians are busy with the intricate wiring of their craft; engineers work on the documentation necessary for each new product; and at one work station, a young engineering associate is testing an auto-ranging to enhance the operation of a Bently Nevada Vector Filter, popular in the industry for diagnostic and balancing work.

"From our standpoint," says Prevette, "the most important factor is capacity, especially Design and Engineering. It is a new way for us to deliver parts and follow, but heavy demands on Engineering and Design could be more than we could handle at this time." However, he acknowledges that such demands were not likely at this stage of development.

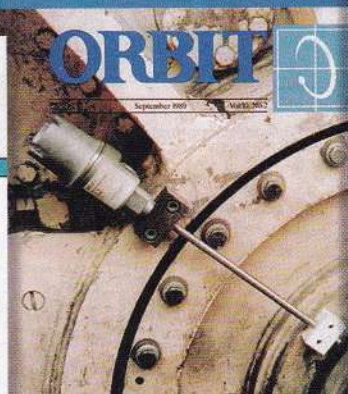
Of all the factors involved in CP's operation, it is a customer relation that occupies Prevette most. "We want the customer to feel that we are there for them, a problem they can count on us. Maybe we can help them by going through our existing product line — maybe the solution is in there, or we could advise them of another method to solve their problem, or we might build something special for them. We want them to tell us their problems and we can start from there."

As for the cost-efficient factors in a department producing a small output when compared to the full production volume of Bently Nevada, Prevette says that Custom Products keeps inventory low.



Jim Prevette, Bently Nevada's Custom Products Manager

BENTLY NEVADA



the Orbit

published by Bently Nevada Corporation

Premier Edition
summer 1980

BENTLY NEVADA INTRODUCES ADRE

Bently Nevada begins the new decade with the introduction of ADRE® — Automated Diagnostics for Rotating Equipment. ADRE is the answer to the tedious, time-consuming chore of acquiring and reducing data collected on the operating parameters of rotating equipment. A product of the resources and experience of our Mechanical Engineering Services group, ADRE presents measured data in a variety of formats to assist in the identification and solution of machine malfunctions.

The basic system includes a desk top computer with CRT and thermal line printer, a spectrum analyzer, a digital vector filter and the necessary software packages. Options include a multichannel tape recorder, multichannel analyzer, floppy disk storage and a real time clock. Data is introduced into the system either from on-line transducer signals or pre-recorded magnetic tape. The resultant signal is played, two channels at a time, into the spectrum analyzer and the DVF's. Outputs are available on the CRT and the hard copy printer in formats including Boof, orbit, polar, spectrum, cosine and balancing.

ADRE can help you realize significant savings and increase plant availability. For a brochure with more information, just check the appropriate line on the attached reply postcard.



WELCOME TO THE ORBIT

Hello! This is the premier edition of The Orbit, Bently Nevada's newsletter for our valued customers. We plan to greet you several times a year with news about our products, and schedules of our trade shows and seminars. The Orbit staff hopes you find the newsletter informative and enjoyable. Write and tell us what you think of our new publication.

NEW LOW SPEED VIBRATION MONITOR

(SEE INSIDE)

ORBIT

FOURTH QUARTER DECEMBER 1999 VOL. 19 NO. 4

Moving beyond machinery protection...

to machinery management

INSIDE YOUR ORBIT:
Using Data Manager 2000 to remotely diagnose gas compressor rotating stall
Interview with the President
Portable data collection and an integrated software platform... two upcoming products that change everything
Integrating critical



HELPING YOU PROTECT AND MANAGE YOUR MACHINERY®

ORBIT

BENTLY NEVADA March 1999 Vol. 18 No. 1

40 Years



Celebrating 40 years of service to industry with reliable machinery protection and management

Today's productivity tool:
Move data not people

ORBIT

January 1986 Vol. 7 No. 1



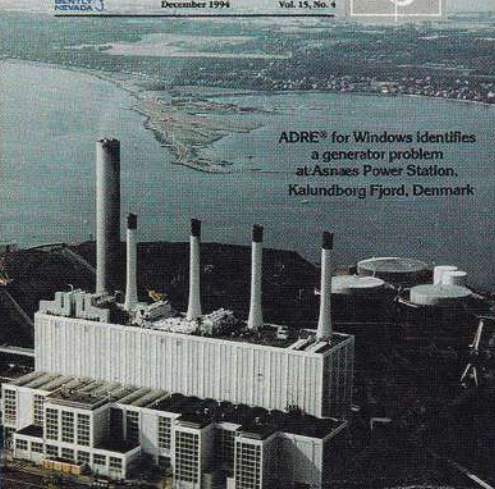
Shaft Crack Detection

ORBIT

December 1994 Vol. 15, No. 4



Celebrating 35 years of service to industry



ADRE® for Windows identifies a generator problem at Asnaes Power Station, Kalundborg Fjord, Denmark



BENTLY Nevada

Looking Back ... Looking Forward ...



Welcome to our 20th anniversary issue! The premiere issue of ORBIT was published in the summer of 1980. It was quite a bit different than those we publish today, but that's not surprising – Bently Nevada itself was different back then. That first issue included this greeting on the front page:

WELCOME TO THE ORBIT

Hello! This is the premiere edition of The Orbit, Bently Nevada's newsletter for our valued customers. We plan to greet you several times a year with news about our products, and schedules of our trade shows and seminars. The Orbit staff hopes you find the newsletter informative and enjoyable. Write and tell us what you think of our new publication.

Today, Bently Nevada is the world's leading provider of machinery management and protection systems and services, and we are especially proud of ORBIT. Over the years, it has evolved from a 4-page newsletter to an extensive technical publication distributed quarterly to over 43,000 people around the world. Yet just as with the first issue, ORBIT continues to go to our "valued customers," and we hope you find it "informative and enjoyable" (the feedback we regularly receive tells us that you do).

In this issue, we spend some time *looking back* at key articles and important messages presented in ORBIT over the past 20 years (page 8), but our focus is *looking forward* to the future as illustrated by the variety of research and new product articles. The article on Dynamic Stiffness (page 18), for example, uses simple physical descriptions, rather than complex mathematics, to present the advantages of externally pressurized fluid-film bearings such as the new Bently ServoFluid™ Control Bearing. The application of an experimental perturbation device produced during the development of the Bently ServoFluid™ Control Bearing is discussed beginning on page 41. There's a discussion of rotating stall in centrifugal compressors from vibrational diagnostics and rotor dynamics standpoints (page 32). On page 26, you will find an article on our new machinery management platform, System 1™, and Snapshot™ for Windows® CE, our latest handheld portable data collection and analysis package. Our new laser-based Field Alignment Kit is presented on page 30.

We gratefully acknowledge our many customers around the world that have contributed to ORBIT, particularly in the form of detailed case histories. Whether authoring articles or allowing us to author material specifically referencing them, customer input has been, and will continue to be, a major factor in ORBIT's success. Of course, we also thank each of you for your ongoing interest in the publication. We are *looking forward* to the next 20 years, and beyond. ☺

– Lane Swensen

ORBIT

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Vol. 21 No. 1

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(4 lb), it is one of the lightest portable data collection tools available, allowing single-handed use for extended periods of time. The removable lithium ion battery provides an *in-use* battery life of at least 8 hours.

Ease of use is a key feature of the Snapshot™ for Windows® CE package. With a clear touch-screen, ¼VGA display, and ergonomically designed keypad, it is simple to operate with either the left or right hand, even while wearing gloves. The backlit display provides convenient viewing of diagnostic plots. In-field analysis and configuration capabilities, along with quick navigation and the ability to view previously collected data, baseline data, and trends, are standard features of this system.

Designed for the industrial environment, Snapshot™ for Windows® CE is also certified for use in hazardous areas with CSA and EExn approvals. The injection-molded Polycarbonate/ABS casing material provides high impact resistance and durability.

A wide variety of measurement types and signal-processing options are available with Snapshot™ for Windows® CE. Large data collection schedules can be accommodated with 24 megabytes of standard on-board memory and extensive in-field analysis capabilities. Support is provided for proximity, REBAM®, velocity, and accelerometer vibration transducers, a temperature measurement option, numeric data, notes, and user-configurable dynamic inputs.

Setting the standard – it's what you expect from Bently Nevada, the world's leading provider of machinery protection and management systems and services. For more information on System 1™ or Snapshot™ for Windows® CE, contact your local Bently Nevada or Bently Nevada (U.K.), Ltd.* sales or service professional. ☺

*Snapshot™ for Windows® CE joins a growing list of products developed and manufactured exclusively by our subsidiary, Bently Nevada (U.K.), Ltd.

Why Do Two Portable Instruments Read Direct Vibration Amplitudes Differently?

This is a commonly asked question. Differences can be attributed to either the sample of data that the measurement is performed on or the way in which the measurement is made.

One of the first things to check is the frequency range over which the direct measurement was performed. For example, consider a signal that has frequency components at 300 and 600 Hz. If a measurement were taken using an instrument with a filter band of 3 to 500 Hz, a direct measurement would not include the 600 Hz component, and therefore would read lower than an instrument with a filter band of 3 to 1000 Hz. Differences in direct filter bands also contribute to measurement differences between portable instruments and permanently installed vibration monitors.

Some portable instruments calculate the direct amplitude from the frequency spectrum. Any frequency component that is outside (usually above) the spectrum frequency band would not be included in the direct measurement. This means that a change in the spectrum sample frequency range can change the direct amplitude reading. If a frequency component exists that is higher in frequency than the spectrum frequency band, the indicated direct amplitude level will be lower than the actual amplitude.

Differences also occur when comparing an instrument that measures zero-to-peak directly (sometimes referred to as true peak) with an instrument that measures root mean square (rms) and then converts to zero-to-peak by multiplying the rms reading by 1.414. This conversion works for a sine wave but does not give accurate readings when converting a complex signal. For a complex signal, a converted rms to peak measurement would read low when compared to a true peak measurement. Bently Nevada instruments measure true zero-to-peak and do not convert rms measurements.

In summary, care should be exercised when using an instrument where the direct amplitude measurement can vary with a change in spectrum plot frequency range or a change in the number of lines of frequency resolution. It is best practice to use an instrument where the direct amplitude measurement is independent of any other data type. All Bently Nevada instruments, including the new Snapshot™ for Windows® CE portable data collector, perform the direct amplitude measurement independently of any other data type configuration. ☺

For an **advanced** approach to your machinery diagnostics,
you need advanced training that combines both
theory and **practice.**



Donald E. Bently
is Founder, Chairman, and
CEO of Bently Nevada
Corporation, and President
of Bently Rotor Dynamics
Research Corporation. Mr.
Bently is actively involved in
machinery dynamics research
& product development. He
received a Master's Degree
in Electrical Engineering –
University of Iowa and an
Honorary Ph.D. – University
of Nevada, Reno.

That's why our Advanced Machinery Dynamics Course is unique. You'll not only learn theory covering advanced rotor dynamic concepts directly from one of the world's foremost authorities – Donald E. Bently – you'll also apply what you learn in hands-on workshops using rotor kits and standard diagnostic tools. The Advanced Machinery Dynamics Course is presented by Bently Rotor Dynamics Research Corporation (BRDRC) and features case histories, demonstrations, hands-on laboratory exercises, and rich discussion opportunities with your peers and our instructors and scientists. It builds on the fundamentals established in our Machinery Diagnostics Course by equipping you to:

- advance your machinery vibration diagnostic skills
- evaluate rotating machinery design and operating parameters
- identify corrective actions
- prevent machinery malfunctions
- create an effective, condition-based machinery management program

If you are a machinery designer, machinery engineer, maintenance engineer, postgraduate student, researcher, or professor involved in rotor dynamics, we invite you to join us for this unparalleled learning opportunity.

Warsaw, Poland, 15-19 May 2000

Registration fee: Early – 1,975 USD (prepaid 4 weeks in advance) Regular – 2,250 USD
Tuition includes: Student course materials, continental breakfast and lunch each day of the course,
one dinner, and one evening reception. Lodging not included. Credit cards are accepted.
To register, contact Bently Nevada Technical Training or your nearest Bently Nevada office.



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1631 Bently Parkway South · Minden, NV 89423
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Website: www.bently.com



...all your machinery



Snapshot™ for Windows® CE

- Surveillance and diagnostic modes – a *single* tool for operators and specialists
- Battery life – 8 hours *minimum*
- Lightweight (approximately 4 lbs!), single-handed operation
- Crystal-clear, backlit LCD display
- Touch-screen user interface – simple to use
- Two full-function channels *plus* Keyphasor® input
- Rugged IP65 sealed case
- Industry-standard Windows® CE operating system
- PCMCIA slot for memory, programming, and hardware connectivity
- Ethernet and serial communications



What if you could take Bently Nevada's online machinery protection and management systems and combine them with portable data collection? What if you could finally manage all your machinery – regardless of how you chose to collect the data – from a single, unified software environment?

You can.

Announcing Snapshot™ for Windows® CE, Bently Nevada's portable data collector.* It's available *now*, and it's destined to be your next portable data collector – a data collector that can be totally integrated with your online system via our System 1™ software. A data collector with the power and features you need to address the hundreds or even thousands of machines in your plant that don't lend themselves to an online system. A data collector that does everything you need – and more – yet at a price that's considerably *less* than what you're probably paying for someone else's non-integrated solution.

Interested? We thought so. To learn more about Snapshot™ for Windows® CE, contact your nearest Bently Nevada sales professional. Snapshot™ for Windows® CE and System 1™ – machinery management will never be the same again.

*Snapshot™ for Windows® CE joins a growing list of products developed and manufactured exclusively by our subsidiary, Bently Nevada (U.K.), Ltd.

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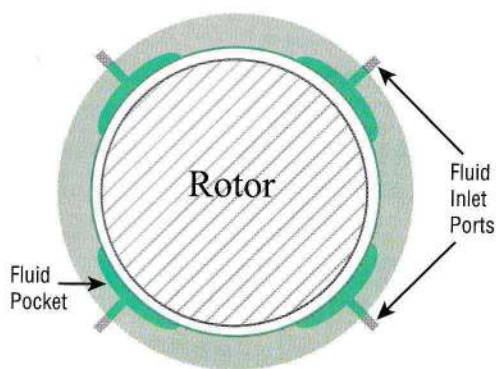
HELPING YOU PROTECT AND MANAGE YOUR MACHINERY®



$$K - M\omega^2 + jD(\omega - \lambda\Omega)$$

(Some see Greek letters...

we see the future of rotating machinery.)



**The ServoFluid™ Control Bearing.
Consider the benefits:**

- Outstanding stability and load capacity
- Excellent stiffness and damping
- Fluid flow allows low operating temperature
- Very low drag & startup torque requirement
- Excellent default mechanisms (simply reverts to a high-pressure or low-pressure fluid-film bearing – no auxiliary bearings required)
- Flexible working fluid options (works with water or with the machine's process media – not just lubricant)
- Suitable for both radial and axial bearing applications
- Ideal for retrofits as a superior alternative to tilting pad bearings

It's called Dynamic Stiffness, and it governs the behavior of every rotating machine ever made. Until now, you had only two choices: you could live with the limitations of conventional bearing designs, or you could pursue actively controlled technologies such as magnetic bearings – a technology with its own limitations.

Not any longer.

Announcing Bently Nevada's ServoFluid™ Control Bearing – an externally pressurized bearing that uses virtually *any* fluid – water, oil, or even the machine's process media itself – to support and control the rotor. A bearing with the familiarity and dependability of simple fluid-based systems, yet dramatically improved performance over conventional designs, suitable for both radial and thrust applications. It can be tailored to the specifics of your application in ways that just weren't possible previously. The opportunities this presents are nothing short of *revolutionary*. Improved performance and improved machine designs are not only possible, they're now *practical* thanks to our ServoFluid™ Control Bearing.

To learn more, visit us on the world wide web at www.bently.com and follow the ServoFluid™ link from our homepage, or contact one of our over 150 sales professionals today. You'll find us in more than 80 locations around the globe.



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HELPING YOU PROTECT AND MANAGE YOUR MACHINERY®

all

culminating in increased Direct Dynamic Stiffness. When a bearing can rely primarily on axial flow of lubricant (rather than tangential) for the support wedge, superior bearing performance results. This permits the design and operation of machinery with much higher inherent stability and better all-around performance characteristics. ☺

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ANNOUNCEMENTS

Donald E. Bently Receives 14th Annual Frederick P. Smarro Award

Donald E. Bently, Founder, Chairman, and Chief Executive Officer of Bently Nevada Corporation, and President of Bently Rotor Dynamics Research Corporation, is co-recipient of the 14th Annual Frederick P. Smarro Award. The award is presented to an engineer who has made a significant contribution to the discipline and practice of plant maintenance and/or engineering. Mr. Charles Bailey, previously of Eastman Chemical Company, is also a co-recipient of the award this year. The Smarro Award was established jointly by The American Society of Mechanical Engineers (Plant Engineering & Maintenance Division) and Reed Exhibition Company in 1987. The award was presented in March at the 2000 National Plant Engineering & Management Show and Conference, in Chicago, which is the venue of National Manufacturing Week. ☺



The 45th ASME Gas Turbine & Aeroengine Technical Congress, Exposition & Users Symposium

**8-11 May 2000
M,O,C, Sports & Fashion Center
Munich, Germany**

Visit us at Stand #275. Among our products, Bently Nevada will be showing a working demonstration of the revolutionary Bently ServoFluid™ Control Bearing.

We will also be presenting the following papers:
Active Controlled Hydrostatic Bearings for a New Generation of Machines, by Donald E. Bently, John W. Grant, and Philip C. Hanifan; *Rotor/Seal Experimental and Analytical Study on Full Annular Rub*, by Donald E. Bently, Paul Goldman, and John J. Yu; *Understanding Machinery Management – Using Machine Condition and Process Information for Maximum Benefit*, by Roger G. Harker; and *Internal Vibration Transducers on an Avon Engine* by Peyton Swan. ☺

2000 Global Training Schedule: April - September

Country	Location	Course	Date
Australia	Sydney	Alignment Fundamentals Workshop	4-6 April
	Sydney	Machinery Diagnostics	29 May - 2 June
	Sydney	Applied Diagnostics Workshop	5-9 June
Canada	Edmonton, Alberta	Getting the Most from Data Manager 2000	11-13 April
	Edmonton, Alberta	Getting the Most from ADRE for Windows	9-11 May
	Toronto, Ontario*	3300 Installation & Maintenance	6-8 June
	Deerhurst, Alberta	Machinery Diagnostics	19-23 June
	Canmore, Alberta	Machinery Diagnostics	14-18 August
	Edmonton, Alberta	Alignment Fundamentals Workshop	29-31 August
	Edmonton, Alberta	Getting the Most from Trendmaster 2000	26-28 September
	Edmonton, Alberta	Getting the Most from Data Manager 2000	4-6 April
Germany	Neu-Isenburg/Frankfurt	Machinery Diagnostics	2-5 May
	Neu-Isenburg/Frankfurt	Transducer Installation & Maintenance	23 May
	Neu-Isenburg/Frankfurt	3500 Installation and Maintenance	24-26 May
	Neu-Isenburg/Frankfurt*	Getting the Most from Trendmaster 2000	27-29 June
	Neu-Isenburg/Frankfurt	Getting the Most from Data Manager 2000	4-6 July
	Neu-Isenburg/Frankfurt	Transducer Installation & Maintenance	18 July
	Neu-Isenburg/Frankfurt*	3500 Installation and Maintenance	19-21 July
	Neu-Isenburg/Frankfurt	Machinery Diagnostics	19-22 September
	Neu-Isenburg/Frankfurt	Applied Diagnostics Workshop	25-28 September
	Neu-Isenburg/Frankfurt	Getting the Most from Data Manager 2000	26-29 September
India	New Delhi	Getting the Most from Data Manager 2000	10-14 April
	Bangalore	Machinery Diagnostics	25-27 April
	New Delhi*	3300 Installation & Maintenance	23-25 May
	New Delhi	Getting the Most from ADRE for Windows	14-15 June
	New Delhi	3500 Installation and Maintenance	28-30 June
	New Delhi	Machinery Diagnostics	19-21 July
	New Delhi	Getting the Most from Data Manager 2000	24-28 July
	New Delhi	Alignment Fundamentals Workshop	9-10 August
	New Delhi*	3300 Installation & Maintenance	29-31 August
	Hyderabad	Machinery Diagnostics	19-21 September
Japan	Tokyo	Machinery Diagnostics	22-25 August
Korea	Seoul	Data Acquisition & Machinery Monitoring	15-19 May
	Seoul	Machinery Diagnostics	22-26 May
	Seoul	Getting the Most from Data Manager 2000	31 May - 2 June
	Seoul	Applied Diagnostics Workshop	28 August - 1 September
Kuwait	Safat	Machinery Diagnostics	19-23 May
Malaysia	Kuala Lumpur*	3300 Installation and Maintenance	4-7 July
	Kuala Lumpur*	3500 Installation and Maintenance	15-18 August
Netherlands	Pijnacker	Getting the Most from ADRE for Windows	18-20 April
	Pijnacker	Data Acquisition & Machinery Monitoring	9-11 May
	Pijnacker	Machinery Diagnostics	15-19 May
	Pijnacker	Getting the Most from Data Manager 2000	23-25 May
	Pijnacker	Applied Diagnostics Workshop	26-30 June
	Pijnacker	Getting the Most from Trendmaster 2000	5-7 September
	Pijnacker	Getting the Most from Data Manager 2000	12-14 September
	Pijnacker	Getting the Most from ADRE for Windows	19-21 September
	Pijnacker	Data Acquisition & Machinery Monitoring	26-28 September
Norway	Bergen	Survey of Machinery Monitoring	29-31 May

Please call your nearest Bently Nevada office for the latest information or contact our Technical Training Department. In the United States, call toll-free 1 (800) 227-5514 ext. 9682. Outside the United States, call 1 (775) 782-3611 or fax 1 (775) 782-9305. Our seminar information is also available on the Internet - www.bently.com.

...all your machinery

(Any questions?)



System 1™ will evolve to include:

- Offline and online data acquisition
- Rolling element bearing support
- Data import and export capabilities using open protocols
- Decision SupportSM connectivity
- Performance monitoring
- Oil analysis
- ...and more!

What if a tool existed that could reduce your costs of training and maintenance for numerous stand-alone software applications, and instead let you manage every machinery asset in your enterprise from a single, powerful program? What if this tool let you build a more extensive asset management system without fear that upgrades and revisions will render your database and integration to other plant systems obsolete? What if this tool let you expand in a modular fashion – adding portable data collection today, and online capabilities in the near future? What if this tool let you add extended functionality with separate components, such as balancing, oil analysis, performance, alignment, document management, Decision SupportSM, and others?

It does.

It's called System 1™, and it will change the way you manage your machinery. That's because you'll be able to concentrate on your machines – *all of them* – from a single, integrated application that's easy to learn, easy to use, and more powerful than anything before it.

Interested? We thought so. Phase I of this revolutionary asset management platform encompasses document management features and support for the new Snapshot™ for Windows® CE portable data collector. To learn more, visit us on the world wide web at www.bently.com/mktreference.htm or contact your nearest Bently Nevada sales professional.

(800) 227-5514 (USA only)
(775) 782-3611

HELPING YOU PROTECT AND MANAGE YOUR MACHINERY®

all


BENTLY
Nevada

Features of System 1™ include:

- Enhanced graphical views of equipment and instrument assets
- Enhanced navigation
- True client/server architecture
- Modular design
- A single configuration utility with features for improved speed and accuracy
- Remote access for off-site analysis
- Rolling element bearing support
- A complete set of diagnostic plot formats
- Advanced alarming and alarm management
- Third-party interfaces
- Document management

Snapshot™ for Windows® CE

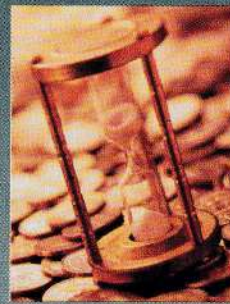
While portable data collection products are not new for Bently Nevada, improving our existing Snapshot™ product line to create Snapshot™ for Windows® CE involved the most extensive user needs definition process of any product Bently Nevada has ever developed. Users were invited to participate in our product definition discussions and exercises that took place in numerous sessions on three continents. These sessions took customer groups through the process of identifying their needs for portable data collection and analysis. These needs were then refined and prioritized by customers in terms of what was most important. The result is a portable data acquisition and diagnostic system that is focused entirely on customer needs.



Snapshot™ for Windows® CE! It's here!

Snapshot™ for Windows® CE is available now and will be completely supported by System 1™. Integration with the host System 1™ software and the ability to interface multiple Snapshot™ for Windows® CE devices provide a cost-effective entrance system into a machinery management platform. The System 1™ platform allows users a path to future integration of offline, periodic online, and continuous online data acquisition with a single display, open database, and many other machinery management tools. Starting with portable data collection, Bently Nevada offers you a means to manage all your machinery assets.

Snapshot™ for Windows® CE features two fully functional channels and a third channel for phase and speed input. It is designed to support the field surveillance needs of both Operations and Maintenance personnel. At less than 1.8 kg



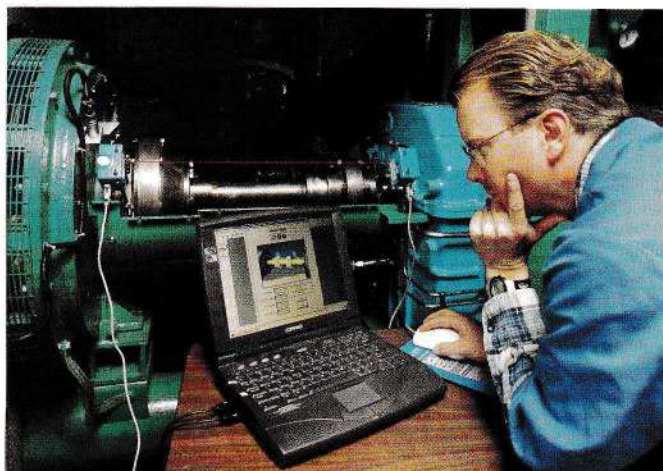
Data Collection: *Time = Money*

Our Vice President of Engineering, Randy Chitwood, recently returned from several weeks of international travel.

Among his activities was a chance to review our new Snapshot™ for Windows® CE portable data collector with customers. Their response to the product? *They liked it so much, they didn't want to give it back!*

They specifically commented on how quickly it obtained readings, compared to the other data collectors they have been using. They estimated the improved speed of data collection meant one less person would be required to collect data; a person who could instead be used to correct (rather than detect) problems. Faster data collection was a feature that was ranked very highly by our customers during the focus groups that were held prior to actual product design. It's gratifying to know that not only did we achieve this goal, but that our customers were quick to translate this improvement in speed to cost savings – a maintenance department capable of doing more work with the same number of employees.

They also remarked about how the product correctly measured direct, unfiltered vibration, in contrast to products they have been using which “derived” this reading (see sidebar on page 28). ☺



Field Alignment Kit in use.

Product features

The Field Alignment Kit provides accurate shaft-to-shaft alignment measurements in a portable kit that is rugged, lightweight, accurate, and easy to use. Product features include:

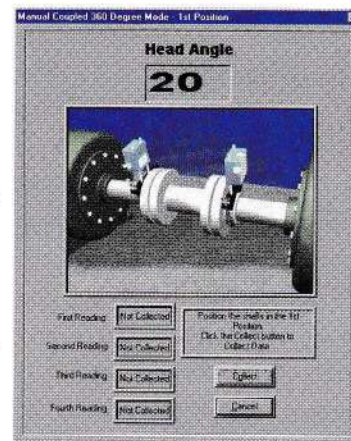
- Coupled and uncoupled shaft alignment
- Only 90 degrees of shaft rotation required
- Manual and automatic data collection modes
- Thermal growth compensation
- Detectors "mapped" for high accuracy
- Each alignment "run" stored in a database and can be deleted or averaged with others

- Alignment tolerance chart capabilities
- Reduced setup time by importing configurations
- Software updates easily downloaded from our corporate website

Product options

It may be desirable to use a Field Alignment Kit in conjunction with several computers. It is also convenient to have the software installed solely for viewing saved alignment databases.

For these reasons, we offer Bently ALIGN™ software with multiple user options: single, three, five, or unlimited users.



Bently ALIGN™ software screen.

It's all about choices

Now the choice is yours – tools or service. Bently Nevada can provide either or both to assure your equipment is aligned properly. Contact your nearest Bently Nevada sales or service professional today to learn more, and be sure to visit our website at www.bently.com for a product data sheet with detailed specifications and additional information on this powerful alignment tool. ☺

ANNOUNCEMENT

Dr. Katsuhiko Ogata Receives the 1999 John R. Ragazzini Education Award

Dr. Ogata, professor of Mechanical Engineering at the University of Minnesota, received the John R. Ragazzini Education Award from the American Automatic Control Council for distinction over a period of thirty years in writing widely recognized system dynamics and controls textbooks. He is the author of seven textbooks, three of which have been published in multiple editions, and most of which were translated into foreign

languages. His most notable book, Modern Control Engineering, is recommended reading for Bently Nevada and Bently Rotor Dynamics Research Corporation engineers. After learning of the award, Donald E. Bently, Founder, Chairman, and CEO of Bently Nevada Corporation, wrote to Dr. Ogata, "The Ragazzini Award is testament to your excellence in communicating and teaching the important subject of control theory." ☺

Country	Location	Course	Date
Philippines	Manila	Transducer Installation & Maintenance	16-18 May
	Manila*	3300 Installation and Maintenance	18-20 July
	Manila	Alignment Fundamentals Workshop	22-24 August
	Manila*	3500 Installation and Maintenance	12-14 September
Poland	Warsaw	Advanced Machinery Dynamics	15-19 May
Qatar	Doha	Getting the Most from Data Manager 2000	2-4 May
	Doha	Applied Diagnostics Workshop	7-11 May
Saudi Arabia	Dammam	Data Acquisition & Machinery Monitoring	1-3 May
	Dammam	Machinery Diagnostics	6-10 May
	Dammam	Machinery Diagnostics	2-6 September
	Dammam	Applied Diagnostics Workshop	9-13 September
	Dammam	Alignment Fundamentals Workshop	16-20 September
Singapore	Singapore*	3300 Installation & Maintenance	4-7 April
	Singapore*	3500 Installation & Maintenance	9-12 May
	Singapore	Machinery Diagnostics	5-9 June
	Singapore	Applied Diagnostics Workshop	12-15 June
United Arab Emirates	Abu Dhabi	Machinery Diagnostics	8-12 April
United Kingdom	Warrington	Getting the Most from Data Manager 2000	11-13 April
	Warrington*	3300 Installation & Maintenance	23-25 May
	Warrington*	3500 Installation & Maintenance	6-8 June
	Warrington	Machinery Diagnostics	19-23 June
	Warrington	Getting the Most from Data Manager 2000	11-13 July
	Warrington	Survey of Machinery Monitoring	5-7 September
	Warrington*	3300 Installation & Maintenance	19-21 September
United States	Houston, Texas*	3300 Installation & Maintenance	4-6 April
	Media, Pennsylvania	Getting the Most from ADRE for Windows	5-7 April
	Media, Pennsylvania	Machinery Diagnostics	10-14 April
	Media, Pennsylvania	Getting the Most from Trendmaster 2000	18-20 April
	Minden, Nevada	Getting the Most from Data Manager 2000	25-27 April
	Houston, Texas	Machinery Diagnostics	1-5 May
	Media, Pennsylvania	Alignment Fundamentals Workshop	2-4 May
	Houston, Texas	Applied Diagnostics Workshop	8-12 May
	Minden, Nevada	Getting the Most from ADRE for Windows	9-11 May
	Houston, Texas	Getting the Most from Data Manager 2000	16-18 May
	Media, Pennsylvania*	3500 Installation & Maintenance	16-18 May
	Media, Pennsylvania*	3300 Installation & Maintenance	23-25 May
	Media, Pennsylvania	Getting the Most from Data Manager 2000	30 May - 1 June
	Houston, Texas*	3500 Installation & Maintenance	6-8 June
	Minden, Nevada	Getting the Most from Data Manager 2000	6-8 June
	Minden, Nevada	Machinery Diagnostics	12-16 June
	Media, Pennsylvania	Getting the Most from ADRE for Windows	21-23 June
	Media, Pennsylvania	Machinery Diagnostics	26-30 June
	Houston, Texas*	3300 Installation & Maintenance	11-13 July
	Media, Pennsylvania*	3500 Installation & Maintenance	11-13 July
	Houston, Texas	Getting the Most from ADRE for Windows	18-20 July
	Media, Pennsylvania	Getting the Most from Data Manager 2000	18-20 July
	Houston, Texas	Getting the Most from Trendmaster 2000	1-3 August
	Minden, Nevada	Machinery Diagnostics	7-11 August
	Minden, Nevada*	3500 Installation & Maintenance	15-17 August
	Houston, Texas	Machinery Diagnostics	14-18 August
	Houston, Texas	Getting the Most from Data Manager 2000	29-31 August
	Houston, Texas	Alignment Fundamentals Workshop	12-14 September
	Media, Pennsylvania*	3500 Installation & Maintenance	12-14 September
	Minden, Nevada*	3300 Installation & Maintenance	19-21 September
	Media, Pennsylvania	Getting the Most from Data Manager 2000	26-28 September
Venezuela	Maracaibo	3500 Installation & Maintenance	8-10 May
	Maracaibo	Machinery Diagnostics	10-14 July
	Maracaibo	Getting the Most from Data Manager 2000	11-13 September

* Optional one extra day for additional topics.

Seminars are usually conducted in the language customarily spoken in the host country.

additional *Interview* with other members of the executive management team provides further insight into where the company is heading in the future and how we will get there.

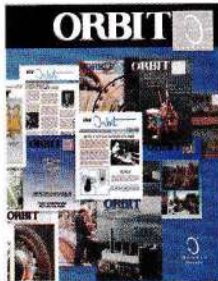
Our Fourth Quarter issue represents one of the biggest evolutions in our history, yet one we are completely comfortable with: adoption of more extensive service offerings, where Bently Nevada assumes considerably expanded scope in protecting and managing machinery. Ideally, we'd highlight every article in the issue; practically speaking, however, you can simply read the article by Steve Riggs and Malcolm Werner titled *Helping You ... A Commitment to Total Engineering Services*. It provides an overview of what to expect from us in the future.

Summary

Twenty years; 62 issues; hundreds of thousands of words. No matter how you view it, the ORBIT magazine is a communication vehicle that Bently Nevada is especially proud of, particularly because you've told us how valuable the articles and content are to you.

For those of you with web access, visit this article online at www.bently.com/articles/400sabin.asp. There, you'll find each of the articles referenced herein is available as a link, allowing easy access to selected historical content of the magazine. For those of you without web access, simply write to us and identify the article(s) by title, year, and issue, and we'll be happy to mail you a copy of the requested items.

In closing, we hope you've enjoyed these first twenty years of the ORBIT as much as we have. May the next twenty years be just as eventful and rewarding. ☺



ANNOUNCEMENT

Enhanced Data Manager® 2000 Machinery Management Software Available

Data Manager® 2000 Software, Version 3.11, released in January. Data Manager® 2000 automatically collects and processes vibration and process data during both steady state and transient (startup and coastdown) machine operation. The new version offers a variety of significant enhancements to help you more effectively manage your machinery, including:

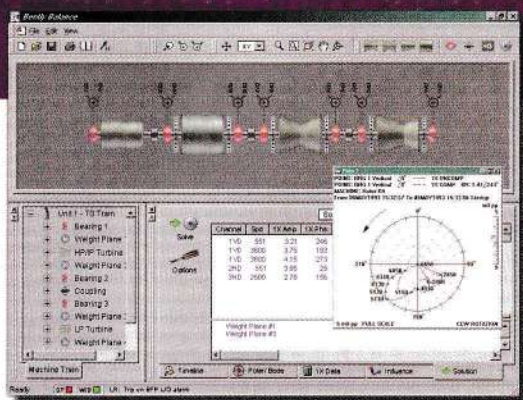
- An OPC Server, add-on module to export vibration data
- Configuration and database auto-archiving
- Deletable specified event files for any point(s) on a machine train
- Auto-uploading of transient buffers
- Faster bar graph updates

- Selectable date/time ranges for acceptance region and shaft centerline plots
- Up to 24 individual frequency labels on a spectrum plot
- Up to four graphical plots printable on a single page
- Selectable printing ranges for the system list, alarm lists, and plot lists
- Active alarm list enabled for archived data
- Support for 3500/45 Valve Position Monitor and 3500/42M Shaft Absolute Monitor

Visit our website at www.bently.com or contact your local Bently Nevada sales or service professional to learn more about this enhanced machinery management tool. ☺

Technically, it's used to **equalize** the synchronous **dynamic forces** caused when a **rotor's geometric** and mass centerlines **don't coincide...**

(but, you can just call it our new balancing software.)



- Multiple correction and measurement plane support
- Calculate, use, and manage multiple Influence Vectors
- Perform graphical "What-if" weight placement solution scenarios
- Supports multi-speed or range-of-speed scenarios, with variable loading
- Automatic data extraction from ADRE® for Windows® databases
- Manual data entry allows the software to be used with any instrument that provides 1X amplitude and phase data
- Full support for Microsoft Windows® 95/98 and NT 4.0
- Historical data and balancing activity retention



HELPING YOU PROTECT AND MANAGE YOUR MACHINERY®

Not ours.

Bently BALANCE™ is a professional tool that makes balancing easy – whether you're dealing with a 50 HP motor/pump set with a single correction plane, or a 500 MW steam turbine generator with a dozen correction planes and complex shaft modes. Bently BALANCE™ is the right tool because it's built on the right theory – the concepts of Dynamic Stiffness using forces, moments, and stiffnesses to model a rotor's behavior. This proven balancing methodology is the key to the power, flexibility, and accuracy of Bently BALANCE™.

Simply put, Bently BALANCE™ is the perfect tool for your balancing needs. To learn more about how it can help you make faster and better balance judgments, visit us on the world wide web at www.bently.com/mktreference.htm or contact one of our over 150 sales professionals today.

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full-time job) at Autonetics, a division of North American Aviation (now Rockwell International) in southern California. There, in a class Mr. Evans was teaching, a student asked what would happen to a typical control system if a certain quadratic approximation broke down. This was the inspiration for Root Locus. Dr. Robert H. Cannon, Jr., Chairman of the Aeronautics and Astronautics Department at Stanford University when Walter Evans was awarded the Rufus Oldenburger Medal, noted, "At North American Aviation's Aerophysics Laboratory where Evans worked, it (the Root-Locus Method) had already become the primary method for designing automatic pilots for high performance aircraft and for the X10 pilotless missile, which was at the time the first operational supersonic aircraft – either piloted or unpiloted."

Don Bently, Founder, Owner, Chairman, and Chief Executive Officer of Bently Nevada, who also worked for John Moore, recalls, "Walt was a most influential person in both my career and in the companies I have founded. While at North American Aviation during the 1950s, it was my privilege to work alongside Walt doing important work on advanced controls for inertial guidance systems. Some of this work was used in the Polaris submarine program that sent U.S. subs beneath the Polar ice cap.... Walt's work on Root-Locus Theory found direct application to much of the work I have been involved with over the past forty-four years, by helping me characterize the behavior of rotating machinery. Walt was a very capable teacher at imparting advanced control theory. In fact, an item I am very proud to have in my personal library is a copy of Walt's book Control-System Dynamics, published in 1954 and signed by

Walt himself. It continues to influence ongoing work today, such as a new fluid bearing that I invented and is now being commercialized."

Walt worked on the technical staff of the Guidance and Control Department of the Re-Entry Systems Operation of the Ford Aeronautic Company from 1959 to 1971, and then returned to Autonetics, where he worked on the technical staff of the Strategic Systems Division until his retirement in 1980.

The Rufus Oldenburger Medal was awarded to him in recognition of his significant contributions and outstanding achievements in the field of automatic control, including his development of the Root-Locus Method, the Spirule (coined as a combination of the words "Spiral" and "Slide Rule"), and his book, Control-System Dynamics.



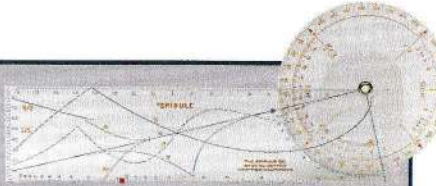
Evans with his sons Greg, left, and Gary, right, in 1972.

Walter was not only a genius of the first degree, he was also a humanitarian. He loved his family (wife, four children, and grandchildren) and his community. He volunteered in the Boy Scouts of America, the United Way, and his church. His daughter Nancy recalls, "He did things differently and would think 'outside the box'; he encouraged us kids to think for ourselves."

His son Greg said, "I remember some of [Dad's] ideas didn't catch on at the time: spray painting tennis balls

bright yellow and orange to improve their visibility, mounting wheels on luggage to enhance their transportability, and staggering work hours to ease traffic congestion. Let's face it, [Dad was] often ahead of the times."

Even after he had a stroke and was partially paralyzed, he was able to teach and encourage others. His daughter Nancy said that he played chess at the senior center and taught the young volunteers to play. When the game was half over, he would turn the board around and play the other side, thus giving them a chance to win. His colleague and friend, Frank Pelton, concludes, "The world has lost a great technological genius. May he rest in peace." ☺



"The Spirule is a plastic device which permits rapid addition of angles or multiplication of lengths of vectors. The Spirule consists of an arm and disk held together with a light friction fit by a special eyelet, which also serves as a pivot point. On a Root-Locus plot, the pivot point is placed at a trial s point and the arm rotated, with respect to the disk, through each of the vector angles, to obtain their sum. A logarithmic spiral curve on the arm permits the logarithm of a vector length to be obtained as an angle, so that the addition of such angles corresponds to the addition of logarithms. The Spirule can also be used in plotting vectors or as a circular slide rule." (Page 237, Control-System Dynamics, McGraw-Hill, 1954.) ☺

In Memoriam

Walter Richard Evans

(15 January 1920 – 10 July 1999)

Walter Evans, brilliant electrical engineer, developer of the Root-Locus Method, author, founder and president of The Spirule Company, inspiring mentor, husband, father, and volunteer, was influential in many people's lives. He had a way of taking complex problems, finding answers to them, and communicating the process to people of more ordinary intellectual ability. The practicality and simplicity of his ideas made Root-Locus Analysis a major advancement in the development of feedback control systems and dynamics systems. Mr. Evans was awarded the prestigious Rufus Oldenburger Medal by the American Society of Mechanical Engineers in 1987, and the Richard E. Bellman Control Heritage Award by the American Automatic Control Council in 1988. Although a genius, he never looked down on others. He sought to teach, instruct, be a model, and serve.



Walter Evans receiving an award at Washington University in 1990.

Evans was born in Saint Louis, Missouri, and his love of math began at an early age. His father was an engineer, and Walt knew when he was young that he wanted to be one, too. He learned to play chess from his grandmother, Eveline Burgess, U.S. Women's Champion for thirty years, 1906-1936, thus enhancing his ability to think a problem through many steps to a solution. Walt's widow, Arline, recalls sitting in front of him in geometry class when they were sophomores in high school, where he would bring in models to prove his answers.

Walt earned his B.S. in Electrical Engineering from Washington University in St. Louis in 1941, completed the three-year Advanced Engineering Training Program at the General Electric Company in 1944, worked as an instructor at Washington University from 1946 to 1948, and obtained his M.S. in Electrical Engineering from the University of California, Los Angeles, in 1951. His thoughts on learning and teaching are reflected in these quotes:

"The main key to learning, in my opinion, is to treat the problem as a game using all the simplifications possible to get the approximate answer."

"...it seems to me the real bulk of learning takes place in self study and problem solving with a lot of positive feedback around that loop. The function of the teacher is to pressure the lazy, inspire the bored, deflate the cocky, encourage the timid, detect and correct individual flaws, and broaden the viewpoint of all."

In 1948, John R. Moore offered Walt a summer job (which turned into a

Root-Locus Theory:

"The roots of a dynamic system's characteristic equation reveal directly and quantitatively the natural behavior it will have – at what frequencies it will vibrate and how quickly the vibrations will damp out. By plotting the locus of the characteristic roots, versus the design parameter being chosen, one can see precisely which values will give good behavior and which will not.... The quick, direct display of behavior characteristics and the powerful, immediate dynamic insight this provides gives Evans' method its central role. Moreover, root loci can be sketched quite accurately in seconds, by eye, using clever sketching rules that Evans developed.... For getting things precise where it is critical to do so, Evans invented the Spirule, with which one can get almost three-significant-figure accuracy, again in seconds.... Thus it is that even today, with great computing power all around us, we typically think about a new control system design in terms of the locus of its roots, which we can sketch in seconds. Then we let our computers work out the details and plot them for us." Dr. Robert H. Cannon, Jr. (speech at the Oldenburger Award Ceremony). ◊

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ANNOUNCEMENT

Bently Rotor Dynamics Research Corporation Appoints New Manager



Paul Goldman, Ph.D., was promoted to Bently Rotor Dynamics Research Corporation (BRDRC) Manager on 15 December 1999. He had been Acting Research Manager for six months prior to his promotion, and, before that, he was a Senior Research Scientist. Paul's new responsibilities include providing major research directions, visions, and plans for BRDRC.

A native of St. Petersburg, Russia, Paul joined BRDRC in 1990. He earned an M.S.M.E. and Ph.D. in Applied Mathematics from State Technical University of St. Petersburg. He has conducted research in rotating machinery dynamics-related topics, including loose pedestal effect, rotor-to-stator rubbing, lateral/torsional cross-coupling, fluid dynamics of oil film bearings, rotating stall, and cracked shaft. He has authored or co-authored over 45 technical papers and reports.

During the experiment, the following peculiarities of the floating ring design were observed:

- The floating ring does not rotate at approximately $\frac{1}{2}$ of the rotor speed. Instead, it rotates much slower – approximately 30 times slower than the rotor.
- Fluid-induced instability produces a forward component of lateral vibrational response with frequency far less than $\frac{1}{2}$ of the journal rotational speed. It corresponds to the whirl of the floating ring with frequency 3 to 4% of the journal rotational speed in the forward direction.
- The floating ring tends to press against the restraints, thus tampering with oil flow. Its “adhesive” ability is so high that at lower journal speeds, the ring does not rotate at all.

Summary

The floating ring adds peculiar features to the bearing. Its centered position between the journal and the bearing is unstable in both axial and lateral directions. It tends to obstruct the oil flow. In some cases, the latter can recover the journal from typical whirl. All the above indicate that using floating rings is a mistake. ☹

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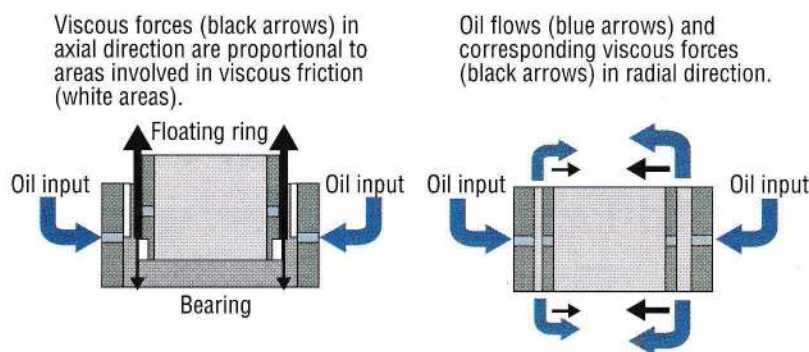


Figure 4. Illustration of axial and lateral instability of the floating ring's centered position.

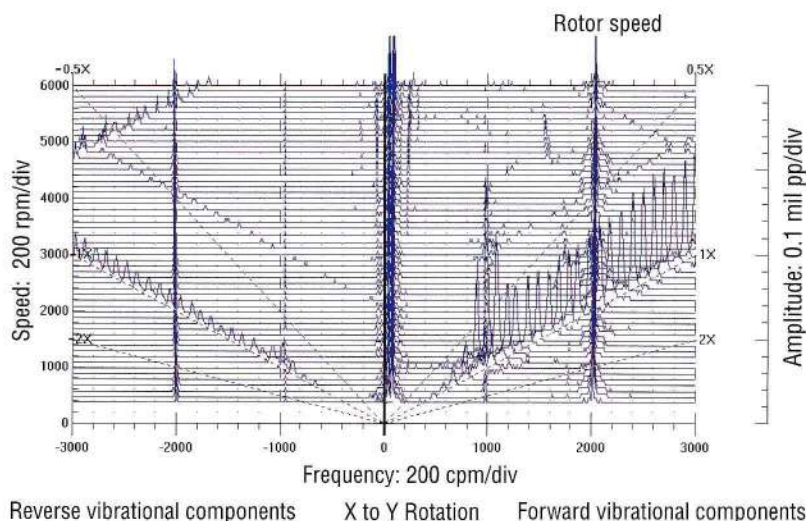


Figure 5. Full spectrum cascade plot of the shaft lateral vibrational response data during startup for the nonsynchronous run of the experimental fluid-film bearing with a floating ring.

ANNOUNCEMENT

Bently Nevada Wins CONTROL Magazine's Readers' Choice Award

The January 2000 issue of CONTROL featured the publication's Readers' Choice Awards for 2000. Once again, CONTROL surveyed its readers and asked them to identify the manufacturers that have provided them with the best instrumentation and control products. No company names were provided to the readers; they had to write in the name of a company. For the sixth year in a row, Bently Nevada placed #1 in the category of Vibration Instrumentation, this year receiving 62% of the votes. CONTROL also asked end-users to rate the customer service of their favorite vendors, and we are very pleased to have been listed as a vendor offering exceptional service. We are grateful to all of our customers that participated in the survey and voted for Bently Nevada. ☹

Adaptation of Perturbation Techniques to Full-Sized Machinery



by Thom Eldridge

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Perturbation testing has characteristically been an analysis tool used to understand the fundamental behavior of rotating machinery and fluid dynamic properties. This terminology and testing methodology should be recognized as a valuable tool in the study of machinery behavior. During the development of the ServoFluid™ Control Bearing (SFCB), Bently Nevada produced an experimental perturbation device (Figure 1) suitable for use on a full-sized machine. The first installation of the perturbator is on a modified Clark 1M6 centrifugal compressor. The perturbator is capable of supplying up to 2,224 N (500 lb) of rotating force at speeds up to 15,000 rpm (250 Hz) which is independent of actual shaft rotation. Through the use of the perturbator, the SFCB development team is able to utilize a mainstay of Bently Rotor Dynamics Research Corporation (BRDRC), the nonsynchronous perturbation of a machine shaft.

Perturbation testing is primarily used to identify a machine's fundamental response with respect to the subject rotor and case system. Dynamic Stiffness properties become available

and are studied by applying a known rotating force with independent frequency to the rotor system by way of a special "perturbation wheel" and measuring the rotor response filtered to the perturbing force frequency. This yields mechanical information about the system's dynamic restraints and stability.

This method of testing allows us to gauge the mechanical response of a turbine, compressor, generator, or pump to a known input force and thus understand the rotor/fluid interactive forces that cannot be identified using synchronous unbalance testing.

Typically, fluid-induced forces in the bearings and seals of a machine are a function of the shaft rotational speed. By holding the running speed constant and varying the perturbation frequency, it is possible to identify the influence of fluid forces. This allows the measurement of the modal behavior of the bearing across all frequencies, instead of only at the frequency at which the shaft turns.

Three classes of perturbation testing

In general, perturbation methodology includes the application of a known force to the structure followed by a measured response that provides

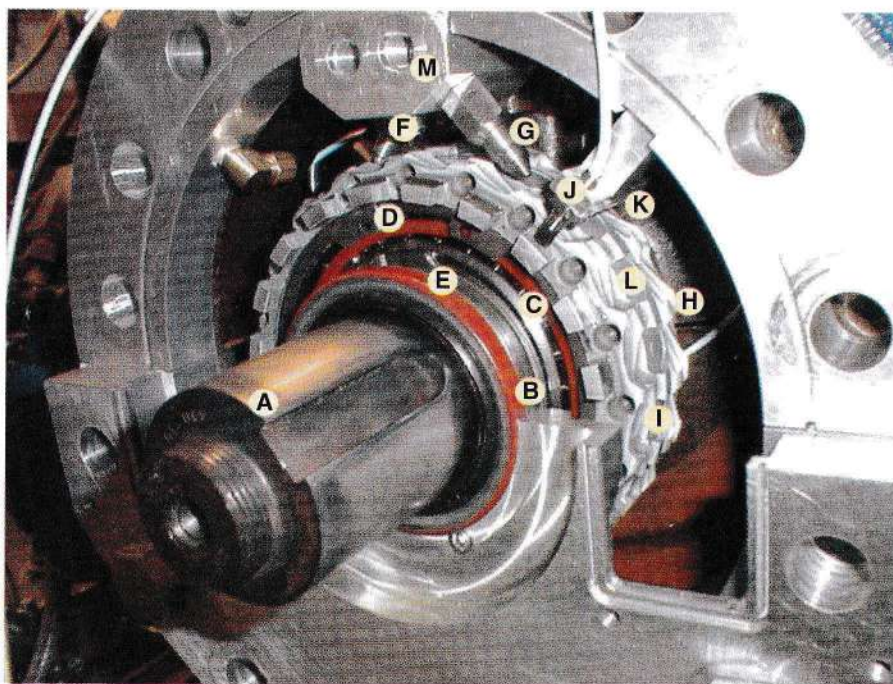


Figure 1. Perturbator assembly: A) shaft; B) adapter shaft sleeve; C) duplex ball bearing; D) external clip ring; E) internal clip ring; F) forward nozzle; G) reverse nozzle; H) forward buckets; I) reverse buckets; J) Keyphasor® transducer; K) speed probe hidden from view; L) speed teeth; and M) manifold.

ω (rpm) measured	F (N) $F = m_u r \omega^2$	Measured Response (mm)		Observed Stiffness (N/m)	
		Direct	Quadrature	Direct $K_D = F / \text{resp}(\text{dir})$	Quadrature $K_Q = F / \text{resp}(\text{quad})$
200	3.29	0.00066	0.0049	4.96×10^5	6.70×10^5
400	13.2	0.0027	0.0098	4.82×10^5	1.34×10^6
600	29.6	0.0064	0.0147	4.61×10^5	2.01×10^6
800	52.6	0.012	0.020	4.30×10^5	2.68×10^6
1000	82.2	0.021	0.025	3.90×10^5	3.35×10^6
1200	118	0.035	0.029	3.42×10^5	4.02×10^6
1400	161	0.057	0.034	2.85×10^5	4.69×10^6
1600	211	0.096	0.039	2.19×10^5	5.36×10^6
1800	266	0.184	0.044	1.45×10^5	6.03×10^6
2000	329	0.536	0.049	6.14×10^5	6.70×10^6

Modal Values		
k	= modal stiffness	= $5000 \frac{\text{kN}}{\text{m}}$
m	= modal mass	= 100 kg
d	= modal damping	= $32 \frac{\text{kN}\cdot\text{s}}{\text{m}}$

m_u	= unbalance mass	= 75 g
r	= radius of unbalance mass	= 0.1 m

Table 1. Example of data collected from perturbation testing.

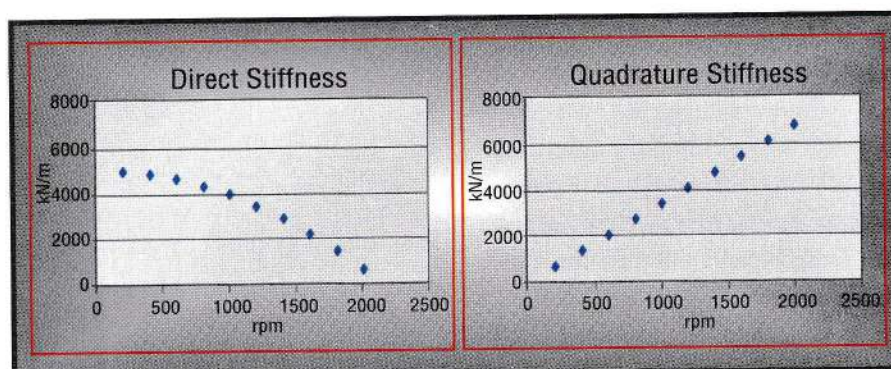


Figure 3. Example of stiffness plots developed from perturbation data.

20-event-per-turn speed signal and is compared to a desired set point. The controller then varies the signal to the valve to achieve the specified speed. While performing a test with the perturbator, a starting speed, ending speed, and acceleration (or ramp-rate) are designated; then the controller smoothly moves the perturbator through the desired range by sending a voltage signal to the flow control valve. Alternatively, if the machine is to be operated at variable or high speed, the controller can be used to hold the perturbator at a fixed speed. Typically, the controller can hold the speed within 5 rpm of a desired value. Additionally, the directional indicator compares the Keyphasor® and 20-event signals to

validate and display the direction of the perturbator rotation.

Summary

Table 1 contains the data collected from a typical perturbation trial, and Figure 3 shows the resulting stiffness plots derived from the perturbation data. By measuring the shaft motion at a progression of perturbation speeds (ω), one can determine the direct and quadrature components of response. The direct response is the motion in the direction of the unbalance force, and the quadrature response is the motion in the direction 90° to the unbalance force (the resultant of the direct and quadrature response is the directly measured phase and amplitude

response). The unbalance force (F) is calculated from the unbalance mass (m_u), radius of unbalance mass (r), and perturbation speed (ω). Direct Stiffness is the unbalance force (F) divided by the direct response, and Quadrature Stiffness is the unbalance force (F) divided by the quadrature response. Finally, Direct and Quadrature Stiffness can be plotted against perturbation speed.

The implementation of perturbation to the Clark IM6 compressor has been highly successful. Bently Nevada engineers are also modifying other machinery to utilize similar techniques. The purpose for perturbing these machines reflects the goals outlined in this article – characterizing the machine and bearings in a safe and reproducible manner. These techniques could be utilized by machine and bearing manufacturers in their design processes or by diagnostic personnel seeking to reproduce or identify a machine malfunction. [Editor's Note: Dynamic Stiffness properties of the new SFCB, determined through perturbation techniques, will be presented in the next issue of ORBIT.] ◊

Bently Nevada's Field Alignment Kit

Full-featured laser alignment



by Dave Saarem

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For over 20 years, Bently Nevada has provided services related to machinery alignment, as detailed in the previous issue of ORBIT. In the article *Bently Nevada's Alignment Services*, we noted that our new Field Alignment Kit would soon be released and that it was already being used by our Global Services organization. Their mission? To confirm that the product meets customers' alignment needs. Is it rugged? Is it easy to use? Does it contain the right features? These questions and others were answered by our service organization. We are now pleased to announce the general availability of the Field Alignment Kit.

With its laser-based hardware and companion Microsoft Windows®-based software called Bently ALIGN™, our Field Alignment Kit provides all the features needed for performing highly accurate alignments, and it is suitable for the rigors of portable field use. The kit is based on the "reverse dial indication" method of performing shaft alignment. To accomplish this, laser/detector heads are mounted on the shaft ends facing each other. The laser from one head is incident on the detector of the other, and vice-versa. When the shafts are rotated together through 360 degrees, the lasers will "migrate" on the detectors. The Field Alignment Kit accurately measures this movement. Knowing this movement and some machine parameters (input by the user), the kit then calculates the machinery misalignment and provides the feedback for correction. The instrument can also calculate misalignment with only 90 degrees of shaft rotation. This is important because more times than not obstructions limit the rotational angle available to the operator. Each laser/detector head is equipped with its own inclinometer so that shaft alignment can be performed on machines that are not coupled together.

Field Alignment Kit:

- Bently ALIGN™ software
- Stationary head
- 3-m (10-ft) serial cable
- 15.2-cm (6-in) slide rod
- 3-m (10-ft) tape measure
- Hardware manual
- Post holder
- Carrying case
- PC card
- Moveable head
- Chain assembly
- 7.6-cm (3-in) slide rod
- V-block assembly
- Quick reference card
- Wrench



information about the structure. The following perturbation classes are distinguished by the type of force (i.e., direction, frequency content, etc.) applied to the rotating machinery.

Synchronous perturbation

Synchronous perturbation relates particularly well to machinery acceptance testing before the machinery is shipped to the plant site. Original equipment manufacturers (OEMs) and end-users must identify the machinery's rotor-synchronous modal mass, stiffness, and damping characteristics of the lower rotor modes. The rotor is in fact being perturbed using an installed "balance" or "calibration" weight (an input force, or forcing function, into the rotor/casing system). The balance ring is typically fixed to the rotor and is therefore considered "synchronous" with rotative speed. At the beginning of a typical balancing project, the machinery is run to establish the "as-found" response. Once the "calibration" weight is installed, the response difference is the rotor system reaction due to that installed weight. The force applied in this case is rotating with the rotor and, therefore, provides a harmonic type of excitation with the frequency equal to the rotational speed. *Synchronous* perturbation does not yield as much information as *nonsynchronous* perturbation; however, it still provides useful data for system identification and malfunction diagnoses.

Unilateral nonsynchronous perturbation (external)

The second type of perturbation testing useful for rotating machinery is the application of a unilateral harmonic force with a sweeping frequency to the machinery case or foundation, in order to identify their stiffnesses and damping characteristics. Typically, whenever

bearing housing, case, or piping resonance is suspected, unilateral perturbation can be used in an attempt to identify these frequencies. In many cases, an application of a *unilateral* force at two major stiffness axes is required.

Rotating nonsynchronous perturbation

Rotating *nonsynchronous* perturbation testing is much more informative since the frequency of the input force is not linked to shaft rotative speed. Perturbation of the machinery can be accomplished by running the perturbator (typically a disk with unbalance and an independent drive system) in the same direction as the rotor direction, otherwise known as *forward perturbation*, or in the reverse direction, known as *reverse perturbation*. Thus a harmonic rotating force is provided with a sweeping frequency that is *independent* of rotational speed of the rotor. Application of this as well as any other perturbation methodology requires the precise knowledge of input forcing frequency, amplitude, and phase, and the response is filtered to the forcing frequency. It must be noted that this type of system may be difficult to set up in the field on typical machinery. The other popular method of perturbing rotating machinery is through impulse testing, which excites many modes simultaneously, but it is not recommended by Bently Nevada because of poor accuracy.

Goals of rotating nonsynchronous perturbation testing

The goals of using a nonsynchronous perturbator on a rotor system are threefold: 1) to identify mechanical characteristics; 2) to determine the modal parameters; and 3) to develop mathematical models leading to understanding of the system stability.

Mechanical characteristics

The perturbator allows us to determine the mechanical characteristics of the rotor system, typically defined by Bently Nevada as Dynamic Stiffness – the direct part of stiffness and the quadrature part of stiffness.

Modal parameters

It is possible to define the modal parameters of a rotor system (stiffness, mass, and damping), along with a number of fluid-dependent parameters (fluidic inertia; λ – the fluid circumferential average velocity ratio; and others), by using simple line and curve fits. The perturbator can be used to determine the linearity of the rotor system. As the unbalance force is increased, the rotor responds at a different dynamic eccentricity within the bearing or seal. By tracking changes in modal parameters as the eccentricity is increased, one can determine the extent of linear behavior of the bearing or seal.

System stability

By utilizing the previously described Direct and Quadrature Stiffness plots, one can define the region of stability. This is particularly helpful when the risks of damage to a machine are high. By using the perturbator to input frequencies at or above the stability margin, it is possible to define this region without operating the machine near instability. Additionally, instability should occur above typical running speed. Without some alternate method of inputting higher frequency forces, it would not be possible to explore these regions.

Rotating nonsynchronous perturbator components

The perturbator developed for use on the Clark 1M6 consists of four major components. The first component

Compressor type	Centrifugal, 1 stage, vaneless diffuser
Working fluid	Air at 84.1 kPa (12.2 psi), 20° C (68° F)
Number of impeller blades	11
Inlet diameter	51 mm (2 in)
Outside diameter of impeller	108 mm (4.3 in)
Diffuser/Impeller radius ratio	1.8
Impeller tip/Diffuser radial clearance	380 μ m (0.015 in)
Impeller/Diffuser axial clearance	640 μ m (0.025 in)
Diffuser/Shroud clearance	3 mm (0.12 in)
Discharge pipe dimensions	51 mm (2 in) inside diameter, 360 mm (14.2 in) length

Table 1. Compressor design and operating parameters.

compressor with a vaneless diffuser, driven by a 373 W ($\frac{1}{2}$ hp) dc electric motor. The motor speed was controlled by a high output, variable power supply, and the speed could be controlled from near zero to 17,000 rpm. The motor was attached to the compressor rotor by a flexible disk coupling.

The compressor rotor was separated into two pieces: a rotating section from the motor to the outboard end of the compressor, and a nonrotating stub shaft from the outboard compressor bearing to the outermost support bearing.

The rotating section was supported by two bearings: a bronze bushing type at the inboard (motor) end, and a rolling element bearing at the outboard end of the compressor. The outer race of the bearing was installed in a collar, which was attached to the nonrotating outboard stub shaft. This allowed free rotation of the compressor rotor relative to the nonrotating stub shaft. A flexible membrane made of stiff rubber supported the stub shaft collar and the bearing assembly. Another bronze bearing supported the outboard end of the nonrotating stub shaft.

An additional 0.8 kg (1.76 lb) disk was attached to the shaft between the inboard bearing and compressor to reduce the rotor system natural frequency.

An orthogonal pair of eddy current displacement transducers was mounted between the air inlet and the impeller to measure the displacement of the rotor.

A spring load frame was attached to the nonrotating stub shaft just outboard of the compressor. This was used to control the radial position of the compressor rotor and to counteract the gravity load of the rotor.

A conical exhaust valve on a lead screw controlled airflow through the compressor. The valve was located at the end of the discharge pipe from the compressor (Figure 2). It could be adjusted through several turns from fully closed, which completely blocked the output flow of the compressor, to

fully open, which allowed free flow with minimum restriction. A pressure transducer measured the compressor discharge pressure.

A constant magnitude, rotating-force perturbator, used in modal testing, was attached to the nonrotating stub shaft. It transmitted an exciting rotating force with variable frequency to the compressor rotor. The perturbator consisted of an outer rotating assembly supported by a rolling element bearing attached to the compressor support structure. Three radial springs with 120° angular spacing were attached from the outer rotating assembly to an inner rolling element bearing attached to the nonrotating stub shaft. One of the springs was tensioned more than the other two springs, and this created a net force in the direction of the most tensioned spring. As the perturbator outer assembly rotated, the springs rotated as well, and a rotating force was created. The force was transmitted through the inner rolling element bearing to the stub shaft, and then to the compressor rotor. A variable speed motor drove the perturbator via a flexible belt. The perturbator drive motor was equipped with a Keyphasor® probe to supply a phase reference and speed indication. The magnitude of the

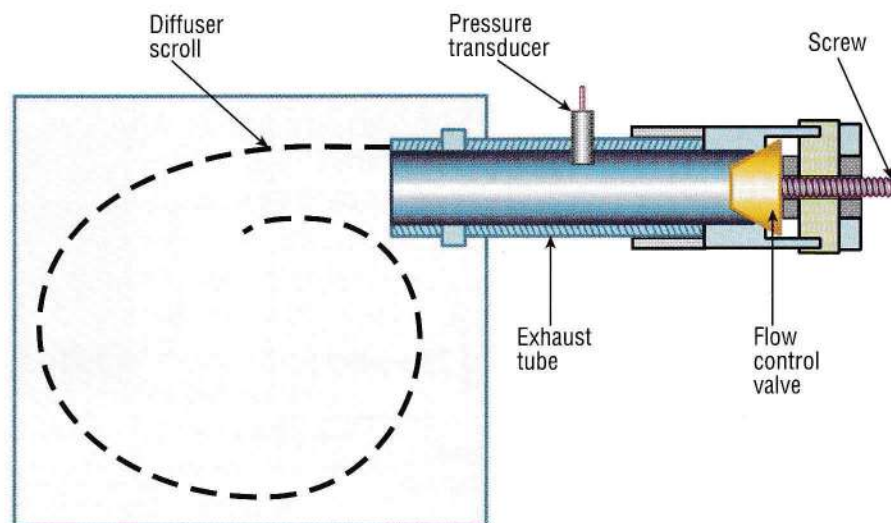


Figure 2. The experimental compressor discharge flow/pressure control.

Tens and Twos ...

The metric system has its opponents and proponents. Proponents argue that a base-10 numbering system simplifies calculations, because many mathematical operations can be accomplished by simply moving a decimal point here or adding a zero there. Simplicity, however, depends entirely on your perspective. Simpler for humans perhaps, but not for computers. With the advent of digital computers and their use of Boolean logic and binary voltage levels for all fundamental computing tasks, I wonder if we wouldn't have been better served by going with a number system that was base-2 (binary), or base-8 (octal), or base-16 (hexadecimal). It certainly would have made us humans more compatible with the computers we've so come to rely upon. If you are a computer programmer, you probably already tend to think in terms of these number systems, and where most of us see "1111" or "F," you see "fifteen." In fact, any number system that has more even divisors is probably friendlier to a computer than the base-10 metric system we've gravitated towards. Maybe our old system of 12 inches to the foot, 16 ounces to the pound, and 4 cups per quart isn't so bad after all ... especially if you're a computer.

Top Dead Center ...

Ever wonder where that term came from? I wasn't able to find out the exact origin (if you know, please let me know), but it likely originated with the steam-fired reciprocating engine in the 1700s (if the piston happened to stop at the exact end of its stroke, the engine would be "dead" when you tried to apply steam again).

Top dead center (TDC) generally speaks of the crankshaft position when a piston is at the very end of its travel in the cylinder, just as it is reversing

direction from the compression part of the engine cycle to the combustion (or power) part of the cycle. Ignition timing is often spoken of in terms of how many degrees it leads or lags top dead center for that cylinder. This makes it similar in concept to a Keyphasor® phase reference, whereby a specific orientation of the shaft (or piston) is used to establish a reference for timing (or phase) relationships.

As an aside, if you really want to "experience" the concept of TDC directly (I don't recommend this), find an antique car with a manual crank to start the engine. However, make sure that the timing is not set so that it fires before TDC. If it does, the crank will "kick back." This phenomenon resulted in many broken or bruised arms before the advent of electric starters.

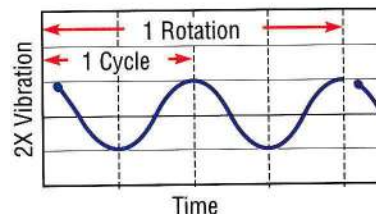
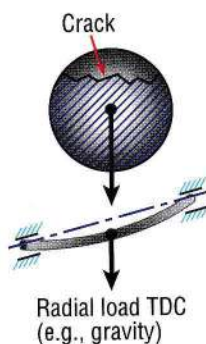
Sometimes you will also hear top dead center referred to as the 12 o'clock position when viewing the circular cross-section of a horizontal shaft, and bottom dead center as the 6 o'clock position. For a horizontal machine, top dead center would then be the vertical direction.

Still others prefer to think of top dead center as the direction in which the radial preload acts, which can be useful nomenclature on both vertically and horizontally oriented machines. However, it does not necessarily correspond to the direction in which gravity acts.

Whatever convention or nomenclature you choose, the concept of "top dead center" is important, particularly when it denotes the direction in which the radial preload force acts on a rotating machine. Even though it is a static

force, it can interact with a shaft to produce a dynamic response.

Consider a cracked shaft, for example. As Figure 1 shows, the crack creates shaft asymmetry. As the shaft rotates, the rotor stiffness changes very rapidly from soft to stiff, twice per revolution. As the rotor changes from soft to stiff under the load from top dead center, the shaft "snaps" in the direction of rotation, creating cyclic reversal stresses. [Editor's Note: For more detailed information, see "Snapping" Torsional Response Can Lead to a Worsening Crack Situation, ORBIT, June 1995.] Some literature has erroneously called the phenomenon a "breathing" crack. From observations we have made, the crack may not open or "breathe" at all. However, it will exhibit this twice-per-revolution change in stiffness. The dynamic result of it — an increase in 2X lateral and torsional rotor responses — is widely used for early rotor crack detection.



Radial load (top dead center - TDC)

1. Shaft snaps in the direction of rotation due to asymmetric stiffness.
2. Cyclic reversal stresses are created.

Figure 1. Shaft "snapping" action and reversal stresses.

The Rule of Holes ...

In closing, I am reminded of this "Rule." It is similar in spirit to Murphy's famous law. The Rule of Holes is simple: If you find yourself in one, stop digging. Our stance on Acceleration Enveloping has recently been criticized by some proponents of this technology. My advice to them? Stop digging. ☺

and coatings are used to reduce the coefficient of friction between the two surfaces. The oil-impregnated bronze bushings used in Bently Nevada rotor kits are examples of dry bearings.

- **Internally Pressurized Bearing** – A bearing in which the rotation of the shaft “drags” the fluid in the direction of rotation to produce a fluid film. These bearings will often have a lubrication supply system that is low-pressure, typically 15 to 20 psi (103 to 138 kPa). The supply pressure is not designed to contribute to the rotor dynamic properties of the bearing in any way; it is simply designed to cause adequate flow through the bearing to provide lubricant cooling (and possibly movement through a strainer or filter to clean the lubricant).
- **Hydrodynamic Bearing** – The most common term for an internally pressurized bearing.
- **Self-Acting Bearing** – A less common term for an internally pressurized bearing. The terms “internally pressurized,” “hydrodynamic,” and “self-acting” are generally used interchangeably.
- **Externally Pressurized Bearing** – A bearing in which the lubricant is supplied at pressures high enough to contribute to, or govern, the rotor dynamic properties of the bearing. Externally pressurized bearings can be further categorized as follows:
 - **Hydrostatic Bearing** – A bearing that does not rely upon rotation of the journal to develop a fluid film; the film is present even with no shaft rotation.
 - **Combination Bearing** – Sometimes also called a “hybrid” bearing, bearings of this type exhibit both hydrostatic and hydrodynamic characteristics. The degree to which the bearing exhibits predominantly hydrostatic or hydrodynamic characteristics depends on the rotational speed and eccentricity at which it is operated.
- **Fully Lubricated Bearing** – A bearing designed to operate with a fully developed fluid film. The fluid may be totally liquid, totally gas, or any homogeneous mixture of the two. While the terms “fully developed fluid film” and “fully lubricated” are often used interchangeably, a fully developed fluid film is really a *characteristic* of a properly operating fully lubricated bearing. Sometimes, these bearings are also referred to as “flooded” or “360 degree” lubrication.

- **Fully Developed Fluid Film** – A condition in which the fluid film fully encircles the journal, with no voids, bubbles, or air pockets in the bearing clearance.
- **Partially Lubricated Bearing** – A bearing designed to have lubricant that does not fully encircle the journal. Sometimes, these bearings are also referred to as “starved” or “180 degree” lubrication.

While the term “hydrodynamic bearing” is generally well-understood, and is widespread in the literature, the terminology when referring to other types of bearings can be somewhat confusing. Also, nearly *all* bearings exhibit both hydrodynamic and hydrostatic behavior, depending on the rotational speed and eccentricity at which they are operated. Thus, the concept of “combination” or “hybrid” tendencies can apply to many bearings. For this reason, Bently Nevada does not prefer to use the term “hybrid” or “combination” when referring to bearings. Instead, we attempt to use the most descriptive terminology possible, such as “externally pressurized fluid-film.” This, for example, is the generic terminology we use when referring to our Bently ServoFluid™ Control Bearing, and it helps to differentiate it from hydrostatic, hydrodynamic, or other bearing types.

Historical Background

Early theories of fluid bearing behavior assumed that the load from the rotating journal was applied directly to the stationary part of the bearing; the role of the fluid was only to decrease friction. Then, at the end of the 19th century, Beauchamp Tower performed a series of precise experiments that measured oil pressure in different parts of the bearing [1, 2]. He came to the following conclusion: Pressure distribution within the oil film is such that the *film* actually exerts the load-carrying forces. Thus, it was the first time a fluid-film bearing was considered a *dynamic system* with certain stiffness properties revealed via oil pressure measurements.

In 1886, a theory by Osborne Reynolds [3] explained these experiments. It produced the famous Reynolds Equation, which described oil pressure distribution in the bearing. The first use of Reynolds’ theory of practical significance was an approximate solution for a specific case of the Reynolds Equation. It was obtained by Fred Ocvirk [4] in 1952 (the so-called Ocvirk solution for an infinitely short bearing). It took more than a century from the time Reynolds originally published his work until a general-case solution for the Reynolds Equation was derived by Bently Rotor Dynamics Research Corporation [5].

By the early 1900s, bearing oil pressure calculations clearly distinguished between full (fully encircling the journal) and partial lubrication. In 1919, W. J. Harrison [6] published work predicting that a fully lubricated bearing would be unstable. This has led to the widely adopted practice of deliberately using partially lubricated rather than fully lubricated designs in the belief that they are more stable. It has also led to the almost universal use of low-pressure bearings, as certain experimental work suggested that pressurizing a fully lubricated bearing would make it more unstable. Recent research, however, contradicts this belief, showing that an *adequately* pressurized and fully lubricated bearing is extraordinarily stable. By adequately pressurized, this means the lubricant is forced to move primarily *along* the shaft (axial flow, i.e., parallel to the shaft), rather than *around* the shaft (circumferential flow, i.e., tangential flow).

While most of the world's standards are currently based on the use of conventional hydrodynamic bearings for turbomachinery, there have been a few notable exceptions. Dr. Judson Swearingen [7], founder of Rotoflow, utilized an externally pressurized bearing design in his line of turbo expanders. Swearingen was an excellent student of rotor dynamic fundamentals. He contributed enormously to the modern design of rotating machinery, although some of his designs are not widely used.

One early documented example of externally pressurized lubrication, relying purely on hydrostatic rather than hydrodynamic principles, was at the Paris Industrial Exhibition in 1878. An exhibit called the "Ice Railroad" consisted of four legs supporting a large mass. The legs rested on a flat steel surface, and oil was pumped down the legs with sufficient pressure to essentially float each leg on a layer of oil. The nearly frictionless mass could then be moved about with ease.

A more modern example of a type of externally pressurized "bearing" that most of us are familiar with is the popular arcade game called "air hockey." The playing table has numerous small holes, closely spaced, through which high-pressure air is forced, allowing the hockey puck to "float," with virtually no friction.

A Bearing Model

A simple spring can be modeled by the familiar equation:

$$\text{Force} = \text{Stiffness} \times \text{Displacement} \quad (1)$$

For a spring, stiffness is a scalar rather than vector quantity, and the displacement is in the direction of the applied force.

For a fluid-film bearing, however, stiffness is a *vector* quantity, consisting of a radial component and a tangential component. We refer to these two components of Dynamic Stiffness as follows:

1. **Direct Dynamic Stiffness.** This stiffness component is in the direction of the applied load.
2. **Quadrature Dynamic Stiffness.** This stiffness component is in the direction *orthogonal* to the applied load.

While our simple spring equation still holds, the stiffness term now has both direct and quadrature components, as does the rotor's response. Therefore, the simple equation (1) is transformed into a vector equation (2):

$$\vec{\text{Force}} = [\text{Direct Dynamic Stiffness} + j (\text{Quadrature Dynamic Stiffness})] \times \vec{\text{Displacement}} \quad (2)$$

For those unfamiliar with the j -operator, it can be thought of as denoting a component that is "90 degrees away" or "perpendicular to" the other component. Hence, the terms "direct" and "quadrature." Mathematically expressed, $j = \sqrt{-1}$. Figure 1 illustrates the direct and quadrature components. When pushing down on a rotating shaft, it not only moves down, *it also moves to the side* (i.e., in the orthogonal direction). Direct and Quadrature Dynamic Stiffness are both important parameters of a fluid-film bearing. Numerous publications ([5, 8-10] represent just four of many) are dedicated to calculating and measuring these parameters. Lambda (λ), the fluid circumferential average velocity ratio, which together with rotational speed and damping determines the fluid drag force [5, 9], is the most important dynamic parameter, closely connected with fluid-induced instability problems [8, 11].

Direct Dynamic Stiffness vs. Quadrature Dynamic Stiffness

In earlier theoretical works ([4], for example) it was erroneously believed that no Direct Dynamic Stiffness exists

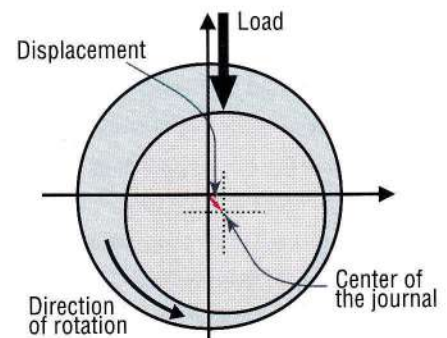


Figure 1. Journal in a fluid-film bearing yields to an applied load in both direct and quadrature directions.

which utilizes both new and existing Bently Nevada products. Progressive testing and evaluation techniques are employed to verify the functionality and compatibility of our new products with our existing products. During these tests, we evaluate the product's ability to meet (and exceed) the design specifications and operating requirements. This testing also identifies areas that may meet the established requirements, but could be improved upon in current or future designs.

One of the main tools used by Quality Assurance, and other departments, is our problem solving methodology. In the early 1990s, we adopted a nine-step problem solving process that revolves around basic quality tools and techniques. This process has been used successfully many times, addressing both internal and external quality issues. The nine steps are:

1. Recognize Expectations (that are not being met)
2. Prioritize
3. Identify Champion
4. Describe the Problem in Detail
5. Short-Term Corrective Action
6. Define and Verify the Fundamental Cause
7. Choose Long-Term Corrective Action
8. Implement Long-Term Corrective Action
9. Prevent Recurrence

To illustrate, we had a situation involving pitting and excess porosity on our 3300 Proximitor® cases. The cases are a purchased part and are manufactured using a metal casting process. A group of 12 employees of varying disciplines and one facilitator was able to use the problem solving process to determine the fundamental cause and corrective action. The problem was random in nature, with no apparent patterns or trends. After going through the data gathering and analysis of the problem, it was evident that the fundamental cause was with the mold used to make the parts. The problem was related to two cavities in a four-cavity mold. The mold was manufactured to Bently Nevada specifications, but had been reconstructed over time to ensure continued quality of parts. Unfortunately, the reconstruction was performed to our assembly drawings, not to the actual component-level part drawings. As such, a portion of the mold (two cavities) did not allow adequate wall thickness of the cast material, thus

allowing pits, cracks, and other cosmetic anomalies to occur. The immediate corrective action was to reconstruct the mold to the appropriate dimensions. The long-term corrective action was to review all of our other molds and their drawings to assure the same problems were not occurring elsewhere. Preventing the recurrence of problems is vitally important.

Have you ever been dissatisfied with the quality/performance of a product? We have a comprehensive complaint and quality reporting process that allows all of our personnel, as well as our customers, to report quality issues whether they are good or bad. Once reported, the problem solving process is used, and appropriate feedback and responses are provided to the originator (if requested). There is even a form on our website (www.bently.com) for submitting quality issues directly to Bently Nevada's Quality Assurance Manager. The input is highly valued, and the information we gain is directly used to make our products and processes even better.

Bently Nevada boasts a comprehensive repair and failure analysis process. We consider each returned product an opportunity to learn and improve. When a product is returned, the nature of the problem and failure mode will dictate the level of analysis performed on that product. For example, our Proximitor® sensors, probes, and cables are not typically considered repairable items. Each one of these products, when returned, is analyzed in our transducer failure analysis lab to make sure we understand the cause of failure and what needs to be done to prevent it from happening again. The information gained from this analysis is then used to improve our new and existing products. It is also used to help us improve our external documentation to ensure that our products are installed and used in a manner that will guarantee continued and lasting use.

Our company-wide quality "assurance" attitude allows Bently Nevada to offer a 3-year service plan on all of our hardware products. This service plan is provided at no additional cost and is available for all of our customers, regardless of their geographical location.

As the concluding line in our Quality Policy states: "We take pride in our ability to design, manufacture, install and service our products in a cost-effective and timely manner." We believe this commitment will continue to keep Bently Nevada on the top of your list as a supplier of high quality products and services that continue to provide lasting value. ○

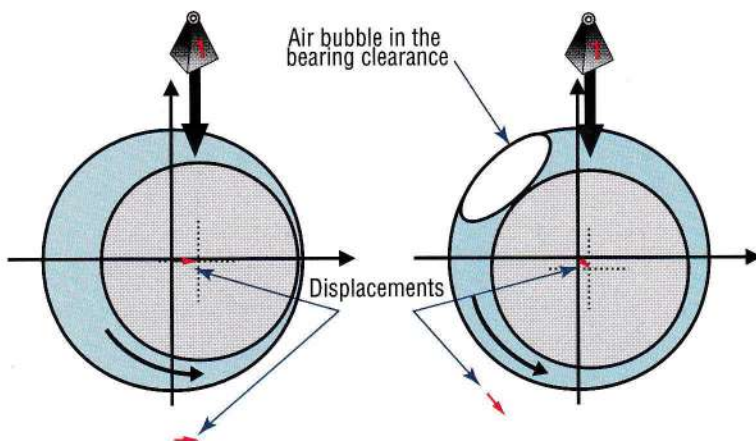


Figure 3. Introduction of an air bubble into the bearing clearance produces a dramatic increase in the Direct Dynamic Stiffness, and alleviates the fluidic inertia effect.

Origin of Direct Dynamic Stiffness

There are certain purely mechanical factors that affect Direct Dynamic Stiffness, such as the stiffness and mass of the rotor itself. The remaining factors, however, are the fluid dynamics of the system.

It was established long ago that the ability of a bearing to carry load is based upon a converging-diverging fluid wedge in the circumferential direction (Figure 4), which exists within a bearing (see [16] for an extended list of references). In the past, this circumferential wedge was considered by researchers as the sole contributor in a fluid-film bearing's ability to carry load. Recently, however, research has shown that fluid flow in the *axial* direction is no less important.

A fluid wedge in the circumferential direction contributes mainly to tangential forces in a bearing, which are sources of Quadrature Dynamic Stiffness [5]. There is, however, *another* wedge contributing to how well the bearing does its

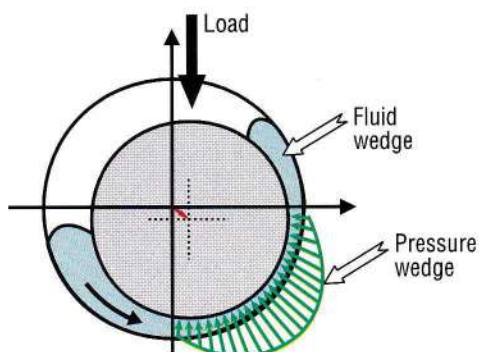


Figure 4. Lubricating fluid forms a fluid wedge in the circumferential direction. Pressure applied from fluid to the journal forms the pressure wedge.

job. This is the pressure wedge (Figure 5) governing fluid flow in the axial direction.

As shown by Bently Rotor Dynamics Research Corporation [5, 9], part of Direct Dynamic Stiffness, created by fluid forces, is proportional to the pressure at which the bearing is supplied with lubricating fluid.

Thus, opposite to the Quadrature Dynamic Stiffness, the Direct Dynamic Stiffness originates from the axial pressure wedge, rather than the circumferential wedge. Flow in the axial direction is determined by the supply pressure, rotative speed, lubricant viscosity, and the pressure drop along the bearing.

Hydrodynamic versus Externally Pressurized, Fully Lubricated Bearings

In hydrodynamic fluid-film bearings, the ability to carry load is mainly produced by means of a hydrodynamic interaction between the bearing and rapidly rotating journal. Circumferential flow of the fluid is usually much faster than the axial flow because the rotating motion of the shaft tends to "drag" the fluid into motion, with speed up to that of the shaft's rotation. This leads to high Quadrature Stiffness. As noted previously, high Quadrature Stiffness without adequate Direct Stiffness leads to instability. Historically, hydrodynamic

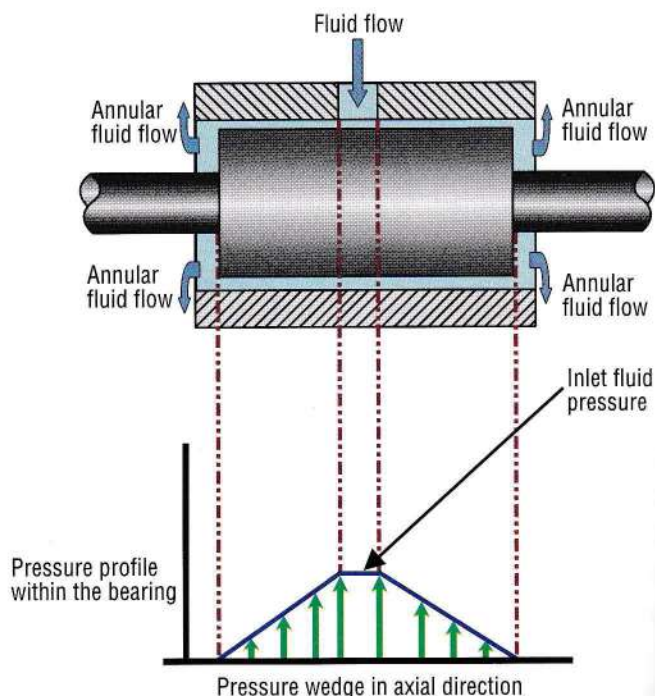


Figure 5. Fluid pressure wedge in the axial direction.

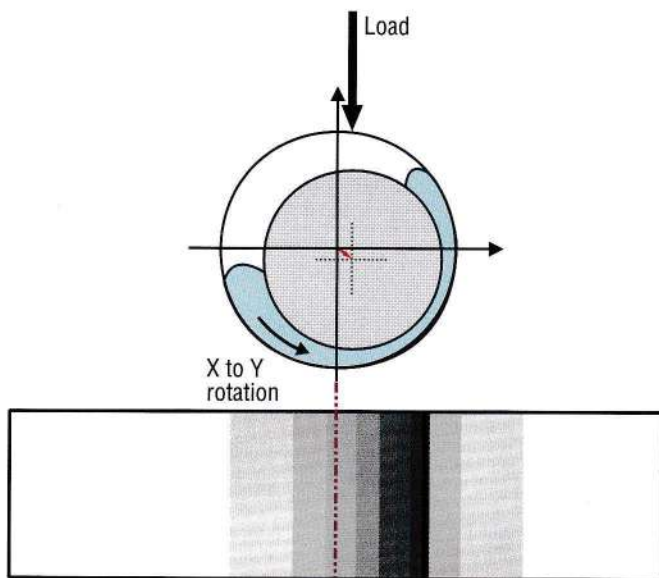


Figure 6. Pressure distribution profile for tangential fluid-film wedge in partially lubricated, low-pressure bearing. Fluid flows primarily *around* the journal. Darker shading denotes higher pressure.

bearing designers have addressed this by increasing the Direct Stiffness as follows:

1. "Starving" the bearing by intentionally keeping it from developing a fully developed fluid film, and relying instead on partially lubricated bearings. The voids introduced in a partially lubricated bearing (Figure 3) increase the Direct Stiffness.
2. Relying on radial preloads (such as gravity on a horizontal machine) to introduce higher Direct Stiffness. Often, machines are even intentionally misaligned to introduce a "stabilizing" preload force.

A better mechanism, however, for increasing the Direct Stiffness is to employ a fully developed fluid film (fully lubricated bearing) and to increase the axial pressure wedge (Figure 5), by increasing the inlet pressure at which the bearing is supplied with lubricating fluid (and hence, an increase in the axial flow). Ironically, this is exactly *opposite* to what is generally taught today. Namely, that externally pressurized bearings should be avoided because they are less stable; and that partially, rather than fully, lubricated bearings are more stable. Bently Nevada's experimental results with a fully lubricated, pressurized bearing (i.e., the Bently ServoFluid™ Control Bearing) show its design to be far more stable than hydrodynamic bearings.

Our conclusion is to use externally pressurized fluid-film bearings, where pressure at the bearing's lubrication ports is

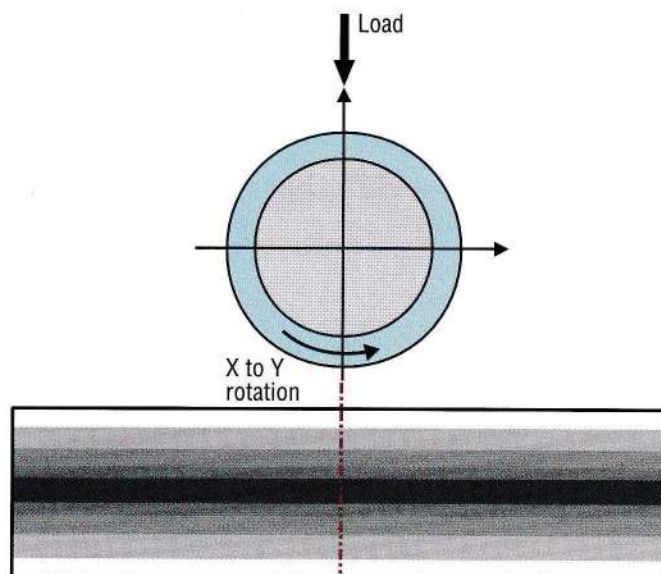


Figure 7. Pressure distribution profile for axial fluid-film wedge in fully lubricated, high-pressure bearing. Fluid flows primarily *axially* along the journal. Darker shading denotes higher pressure.

high enough to create even radial flow and dramatically increased axial flow. Figures 6 and 7 illustrate the difference in pressure profiles for bearings with primarily tangential flow versus those with primarily axial flow.

Summary

In hydrodynamic fluid-film bearings, the ability to carry load is mainly produced by means of the hydrodynamic interaction between the bearing and its rapidly rotating journal. There are only two directions of considerable fluid flow: circumferential and axial. Fluid flow in the circumferential direction creates a fluid wedge with pressure varying greatly in this direction. When a fluid film is fully developed (no voids), the circumferential wedge produces only Quadrature Dynamic Stiffness.

By supplying adequate external pressurization to such a fully lubricated bearing, a sufficient pressure wedge in the axial direction can be produced, which provides axial flow. This axial pressure wedge contributes to Direct Dynamic Stiffness of the bearing. Direct Dynamic Stiffness not only helps the load carrying capacity of the bearing, but also helps to solve stability problems, which are quite common in fluid-handling machinery.

While currently widely employed, "bearing starvation" is a poor and dangerous way to influence stability. Instead, the use of externally pressurized fluid-film bearings, such as the Bently ServoFluid™ Control Bearing, leads to increased axial flow of lubricant, which improves the axial pressure wedge,

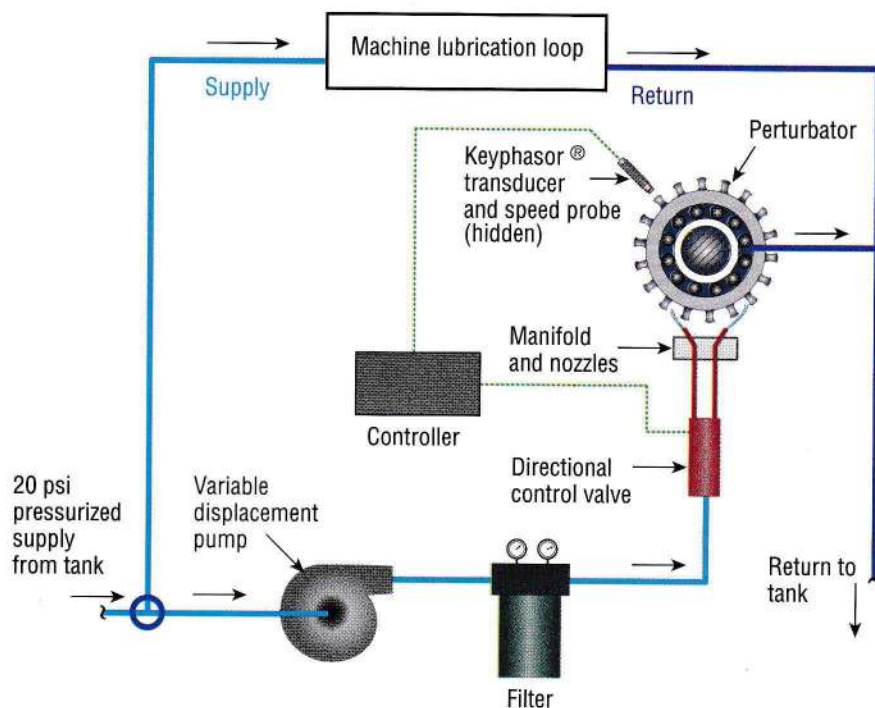


Figure 2. Schematic of fluid delivery system/perturbator showing the necessary pumping, filtering, and regulating elements required to supply the perturbator directional valve with fluid. The perturbator and machine lubrication systems share a common cooling loop, drain, and reservoir.

is the “wheel” of the perturbator. This 7.62-cm (3-in) long, 15.24-cm (6-in) diameter cylinder has two rings of teeth, two counter-directional sets of buckets, and a series of threaded holes. Counter-facing buckets allow the perturbator to operate as a Pelton wheel. When a high velocity stream of fluid is directed at one set of buckets, the perturbator turns in the forward direction; it turns in the reverse direction when the other set of buckets is used. The rings of teeth serve two purposes: 1) to supply a Keyphasor® signal from a distinct tooth for use with ADRE® for Windows® (Bently Nevada’s data acquisition platform); and 2) to supply a 20-event-per-turn speed signal for use with a digital tachometer, electronic speed controller, and directional indicator. Threaded holes allow an unbalance weight to be placed at a given radius and angle, which produces a known force

when the perturbator turns. The force magnitude is described by $F = m_u r \omega^2$, where “ m_u ” is the mass of the unbalance, “ r ” is the radius of the unbalance mass, and “ ω ” is the rotational speed of the perturbator. Using fluid to drive the perturbator avoids numerous undesirable effects, such as a separate drive system, side loading from belts or chains, and difficult access through the machine casing.

The second component is a duplex ball bearing that allows the perturbator wheel to spin independently on the shaft and transmit the unbalance force to the shaft. The two parts of the duplex bearing are held together with an external clip and locknut attached to a sleeve placed on the shaft. This sleeve serves to adapt a metric-series bearing to an English-sized shaft. Similarly, a pair of internal clips is placed at the inner diameter of the perturbator wheel

to hold the perturbator axially along the duplex bearing.

The third component is the fluid delivery system (Figure 2): a pressure-compensated, variable displacement piston pump; a three-way, proportional, flow-control valve; a manifold; and two nozzles. These components are located external to the machine except for the internally mounted manifold and nozzles, which are connected via high-pressure hoses and fittings through the machine case. The piston pump is mounted in a “piggy-back” manner with the main fluid supply pump used to pressurize the ServoFluid™ Control Bearing. The pump is capable of supplying up to 18.9 L (5 gal) per minute at 2.07×10^7 Pa (3000 psi). The three-way control valve has a normally closed configuration. When the electrical control signal shifts the valve toward either port, high-pressure fluid passes to the manifold where it exits through the appropriate nozzle. Fluid exiting the nozzle moves at high velocity (upwards of 152 m/sec, or 500 ft/sec) toward the perturbator wheel. This high-velocity fluid carries kinetic energy, which can be transferred to the perturbator wheel in the form of rotational energy. Speed of the perturbator can be controlled by opening the valve more, or less, in order to vary the speed of the fluid stream. In order to slow or reverse the direction of rotation, the valve is shifted, sending fluid to the opposite nozzle where the fluid stream strikes the buckets facing the opposing direction. This allows the perturbator to perturb the machine in either the forward (same as rotational direction) or reverse (opposite rotational direction) orientation.

The final component is a controller, which sets the direction, speed, and acceleration rate of the perturbator. Perturbator speed is measured using the

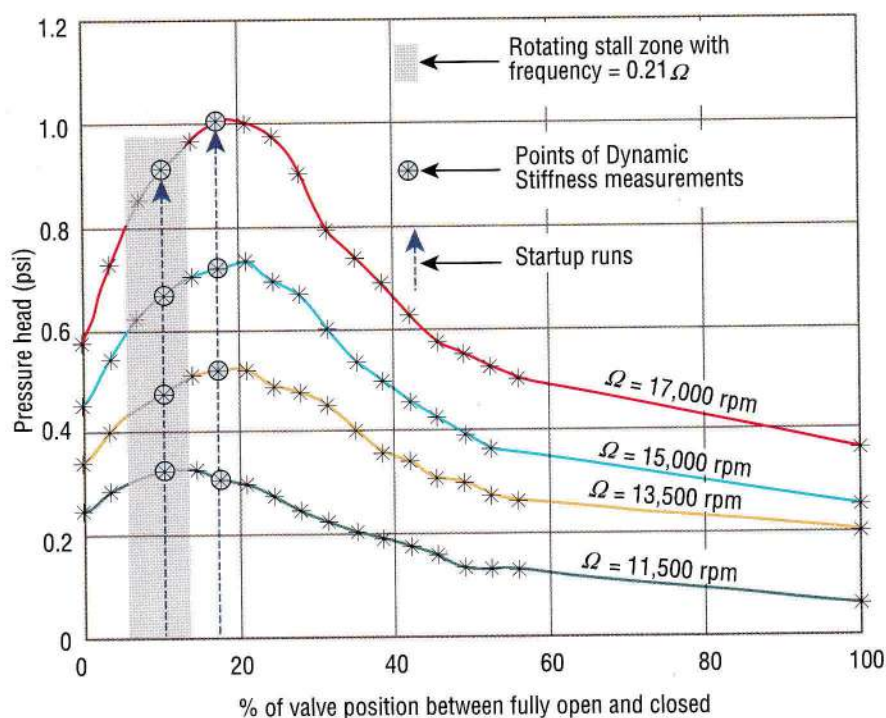


Figure 3. Experimental compressor performance chart. Ω = rotative speed.

rotating force, 1.3 N (0.3 lb), did not change noticeably with speed, and the perturbator provided a variable speed, constant magnitude, rotating-force excitation to the compressor rotor.

An experimental compressor performance chart is shown in Figure 3. It also depicts the points on the performance curves where the vibrational data was taken.

The first series of experiments consisted of compressor startups with two different discharge valve positions, while the perturbator was at rest. Valve positions, which were as close to each other as possible, were chosen to provide normal operating conditions and rotating stall conditions. During normal operation, predominantly 1X forward vibration was observed, with no

subsynchronous vibration. The data is presented in the form of a 1X, forward component Bode plot and full spectrum cascade plot (Figure 4). The Bode plot shows the existence of two rotor modes in the range from 0 to 17,000 rpm. With the help of an additional set of probes, the first mode was identified as an almost rigid body conical mode of the shaft, while the second was a bending mode. The data taken during rotating stall is shown in Figure 5. Since the Bode plot of the 1X forward component in this case does not differ strongly from the previous experiment, only a typical orbit and full spectrum cascade are presented. The experimental compressor rotor exhibits rotating stall whirl with a subsynchronous frequency component of 0.21X in the forward direction. The subsynchronous component of the compressor rotor radial vibration is most likely caused by the diffuser rotating stall.

Rotor dynamic characteristics of rotating stall

The startup data does not provide sufficient information to identify any changes in the Dynamic Stiffness of the first mode due to rotating stall, because the mode occurs at rotative

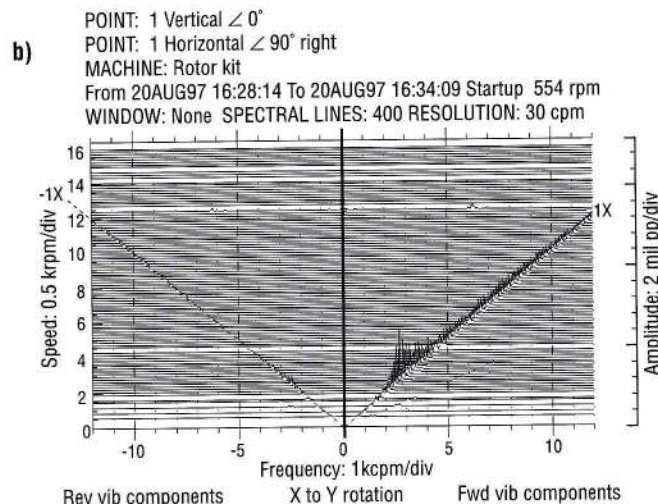
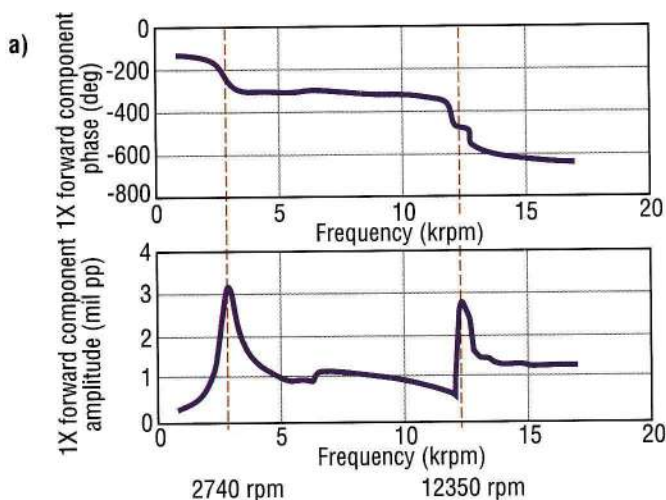


Figure 4. Rotor lateral X and Y vibration data of the experimental compressor during startup without rotating stall: a) 1X forward component Bode plot, b) Full spectrum cascade plot.

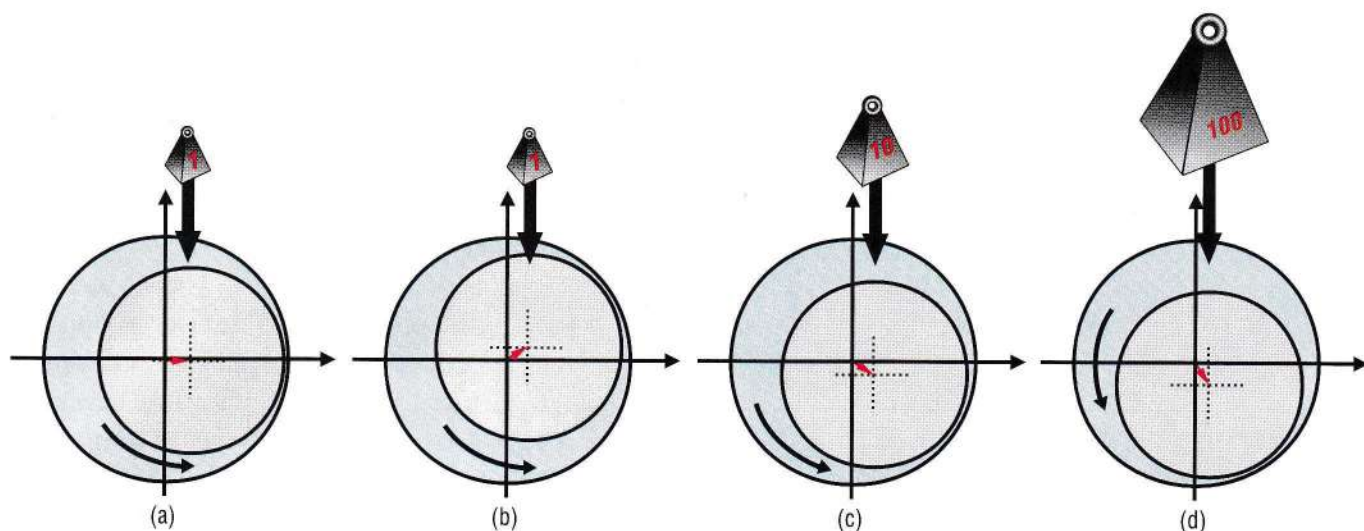


Figure 2. For a fully developed fluid film, the heavier the load, the more the journal's displacement direction coincides with the applied load's direction.

in a fluid-film journal bearing when there is a fully developed fluid film. However, this belief contradicted experimental data, which frequently showed behavior of a fluid-film journal bearing under load as shown in Figure 2.

Under a light load (Figure 2a), the journal displacement is almost perpendicular to the direction of the load, which indicates that the total Dynamic Stiffness is dominated by the quadrature, rather than direct, term. That is, the rotor moves perpendicular to the applied force. When fluidic inertia effects are present, lightly loaded rotors may even move *above* the bearing centerline (Figure 2b). With increasing load (Figure 2c), the rotor begins to move in the direction of the applied load. Finally, under very heavy load (Figure 2d), the Dynamic Stiffness is dominated by the direct, rather than quadrature, component. This yields displacements primarily in the direction of the applied load.

Partially Lubricated Bearings, Direct Dynamic Stiffness, and Stability

When the fluid film is not fully developed, Direct Dynamic Stiffness increases dramatically and eliminates the fluidic inertia effect. This results in a change of both direction and magnitude of displacement: the direction becomes closer to that of the applied load and the magnitude decreases, since total Dynamic Stiffness (including both direct and quadrature components) becomes larger (see Figure 3).

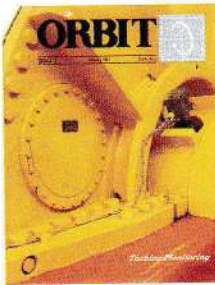
Direct Dynamic Stiffness plays an important role in the problem of stability: Stability requires that the Direct Dynamic Stiffness be of the same order of magnitude as the

Quadrature Dynamic Stiffness. As we have shown, introduction of some void into the circumferential flow does increase the Direct Dynamic Stiffness. Unfortunately, it was erroneously concluded that because introduction of a void (partially lubricated bearing) increased the Direct Stiffness of a bearing, it was an effective way to improve stability. Later such an approach was shown to be misguided. Starvation of a bearing (which is just another term for introducing a void) introduces a new, dangerous type of instability [12], which sometimes leads to catastrophic failures [13-15]. Such failures can occur when the rotor is in a classic fluid whirl condition, where the whirling frequency is about 47% of shaft rotational speed (0.47X). This causes the journal to act like a pump – pumping the lubricant out of the bearing. Under such conditions, lubricant flows not only through the usual drain openings, but also through the supply pipes, and in the case of low lubricant supply pressure, results in gradual acceleration of lubricant starvation. This can ultimately result in the transformation of the whirling frequency from 0.47X to 0.33X, which produces a wide band of other frequency components. The latter whirling frequency leads to dramatic changes in local temperature, and thinning of the lubricant, further worsening lubrication conditions. The final result may be as dramatic as charring of lubricant in lubricant passages [14].

Instead, the proper way to improve Direct Dynamic Stiffness is by increasing the pressure drop *along* the bearing. Following are the basic principles of Direct Dynamic Stiffness, as well as major factors affecting it.

enhanced capability for Bently Nevada's Global Services organization with new computerized analysis tools developed through Bently Rotor Dynamics Research Corporation (BRDRC). Numerous articles also appear related to the need for, and economic justification of, machinery information for all classes of machinery (not just critical machines) – evidence that the “little red all” added to our mission statement is not a new concept.

October's issue finds Mr. Bently publishing *More About That Damping Term* with a further discussion of λ and its importance. The article is continued in the first issue of 1987 in the *Bently's Corner* column.



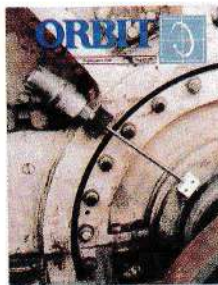
1987

Our investigation into more effective methods for addressing rolling element bearing machinery continues in the August issue with *Predictive Maintenance Through the Monitoring and Diagnostics of Rolling Element Bearings*. Enhanced abilities to detect shaft crack are detailed in *Shaft Crack Detection Using the Acceptance Region*, a software plot now available in all Bently Nevada programs. Finally, the August issue also contains an important “Back-to-Basics” article called *Axial Thrust Position Monitoring – Part 1*. (Part 2 appears in the April 1989 issue.) It's worthwhile reading regarding the importance of this critical machinery measurement and proper instrumentation adjustments to measure it accurately.



1988

Our 3300 Series Monitoring System is introduced to the world in the September issue, poising Bently Nevada for a decade of continued growth as customers rapidly adopt this new product for virtually all machinery protection applications.



1989

Important work by Don Bently and Agnes Muszynska on *Fluid-Generated Instabilities of Rotors* is published in the April issue, summarizing their numerous (over 25) technical papers on the subject.

In the December issue, our Trendmaster® 2000 system is introduced, offering an online alternative to portable data collection for general purpose and essential machinery. In retrospect, this product was probably a decade ahead of its time – we are just now beginning to see significant interest in online systems for this class of machinery, as industry's new rallying cry becomes “automated asset management for all machinery.”



1990

In the June issue, Bently Nevada introduces a digital Serial Interface option for our 3300 Series Monitoring System, marking the start of a new era for interfacing to process control systems relying on digital, rather than analog (e.g., 4-20 mA), methods. This is an important milestone in our journey towards providing information to operations personnel, not just machinery specialists.

Don Bently and Malcolm Werner offer some good advice on basic rotor design considerations in their article *Extending Machinery Life*, in the September issue.

In December, we devote a significant part of the magazine to monitoring for reciprocating compressors, marking the start of our commitment to provide solutions for these machines.

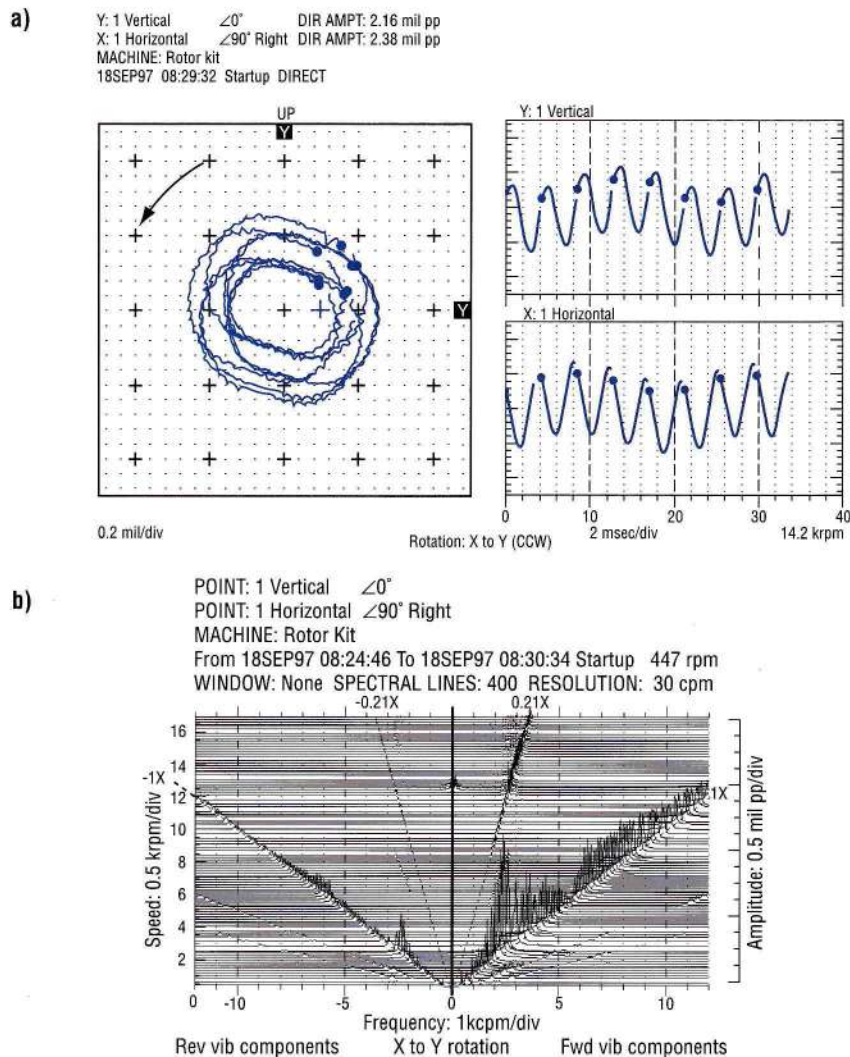


Figure 5. Rotor lateral X and Y vibration data of the experimental compressor rotor during startup with rotating stall: a) Typical orbit/timebase, b) Full spectrum cascade.

speed below the onset of rotating stall whirl. A series of nonsynchronous perturbation tests were performed at each of four compressor running speeds: 11,500; 13,500; 15,000; and 17,000 rpm. At each running speed, perturbation trials were performed for the two valve positions used in the startup tests. The constant force perturber generates a force of magnitude F with orientation δ rotating in the same direction as the compressor rotation (forward perturbation), resulting in the perturbation vector $Fe^{j\delta}$. The perturber was ramped up from slow roll speed to 3,000 rpm in order to identify Dynamic

Stiffness of the rotor first mode. The vibration signals from two orthogonal eddy current displacement transducers, located at 90° to the right (X) and 0° (Y) relative to vertical (as seen from the driver), were filtered to the frequency ω of the perturbation force. This yielded amplitudes and phases for vertical and horizontal components of the response.

In order to eliminate the minor influence of mechanical stiffness asymmetry, 1X vertical $\bar{Y}_\omega$ and horizontal $\bar{X}_\omega$ vectors were combined into the circular component:

$$\bar{A}_\omega^{(fwd)} = \bar{Y}_\omega - j\bar{X}_\omega \quad (1)$$

Next, the perturbation process was repeated for perturbation in a direction opposite the rotation of the compressor rotor (reverse perturbation) for each running speed Ω and valve setting. In this case, the forward (in the direction of perturbation) component of the filtered-to-perturbation-frequency response is opposite to the direction of rotation.

The nonsynchronous Dynamic Stiffness was calculated by taking the ratio of the force input to the response vectors for each speed in the databases.

$$DS(\omega) = \frac{Fe^{j\delta}}{\bar{A}_\omega^{(fwd)}} \quad (2)$$

$$DDS(\omega) = \text{direct}(DS) \quad (3)$$

$$QDS(\omega) = \text{quad}(DS) \quad (4)$$

Since the total Dynamic Stiffness of the compressor consists of a mechanical and an aerodynamic part, the former was measured by running nonsynchronous perturbation on the compressor at zero rotative speed (DS_0). The difference between the Dynamic Stiffness of the running compressor and that of the stopped compressor represents aerodynamically-induced Dynamic Stiffness, DS_{air} :

$$DDS_{air}(\omega) = \text{direct}(DS - DS_0) \quad (5)$$

$$= K_{air} - M_{air1}\omega^2 - 2M_{air2}\Omega\omega$$

$$QDS_{air}(\omega) = \text{quad}(DS - DS_0) \quad (6)$$

$$= D_{air}\omega - T$$

where M_{air1} , K_{air} , and D_{air} are air inertia, radial stiffness, and damping, respectively, generated by the air/impeller interaction, M_{air2} is a fluidic mass term, and T represents a tangential force coefficient.

Equations 5 and 6 show the parameters of Direct and Quadrature Stiffness extrapolation, that fit the data within an accuracy of about 2% compared to actual.

of the floating ring at high speeds (Figure 2). It resulted in lubricant starvation, thus in the whirl disappearance.

In order to clarify the observed phenomena, an experimental rig with a floating ring bearing was designed and built.

Experimental study of a fluid-film bearing with a floating ring

A vertical rotor consisting of a shaft and a journal, supported inboard by a bronze oilite bushing and outboard by a cylindrical oil-lubricated bearing with the floating ring, was driven by a 74.6 W (0.1 hp) electric motor (equipped with a speed controller) through a flexible coupling (Figure 3). A free spinner with a 2.4 g (0.08 oz) unbalance mass was installed next to the outboard bearing. The free spinner was driven by a separate 74.6 W (0.1 hp) electric motor (with a speed controller) through a flexible coupling. Both the rotor and the floating ring lateral motions were observed by sets of proximity probes in XY configuration. Three additional transducers were used to observe phases of the rotor, floating ring, and free spinner. Diametral clearances between the bearing/ring and journal/ring surfaces were 2.54×10^{-4} m (10 mils) each. Length and diameter of the shaft were 0.30 m (12 in) and 0.0095 m (0.375 in), respectively.

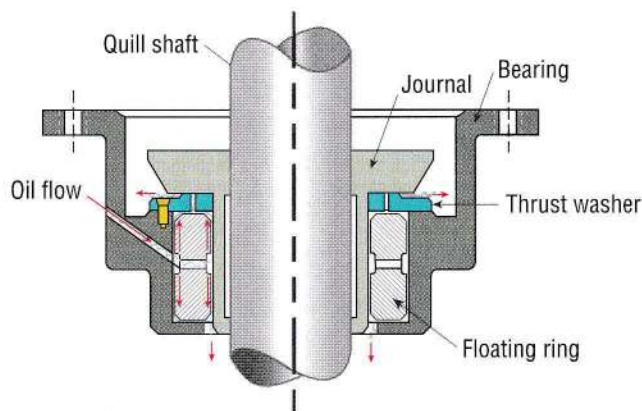


Figure 2. Journal/thrust combination.

Length and diameter of the journal were 0.038 m (1.5 in) and 0.029 m (1.125 in), respectively. Length of the floating ring was 0.025 m (1 in) and inner diameter of the bearing was 0.064 m (2.5 in).

The fluid-lubricated bearing was radially supplied with T10 oil from four laterally and axially symmetric ports. There were two circumferential grooves on the inner and outer sides of the floating ring and four symmetric radial holes to supply oil to the clearance between the journal/ring surfaces. The oil outlet was located at the upper axial side of the bearing. Masses of the shaft, journal, and floating ring were 0.17 kg (0.37 lb), 0.26 kg (0.57 lb), and 0.40 kg (0.88 lb), respectively. The hydrostatic oil pressure at the inlet was 13.8×10^3 Pa (2 psi).

Figure 4 illustrates axial and lateral instability of the centered position of

the floating ring. In either case, the fact of instability is determined by definition: If the ring is initially slightly displaced from the centered position in between the bearing and the journal in either axial or lateral direction, then oil flow will tend to increase the initial displacement.

The results of nonsynchronous perturbation of the rotor are presented below. During that testing, the unbalanced free spinner rotates with swept frequency while speed of the rotor is constant.

The full spectrum cascade plot of the shaft lateral vibrational response data during startup for the nonsynchronous run is presented in Figure 5. During the nonsynchronous run, the rotor of the experimental rig rotated at the constant speed of 2000 rpm, while the unbalanced free spinner was accelerating from 0 to 6000 rpm.

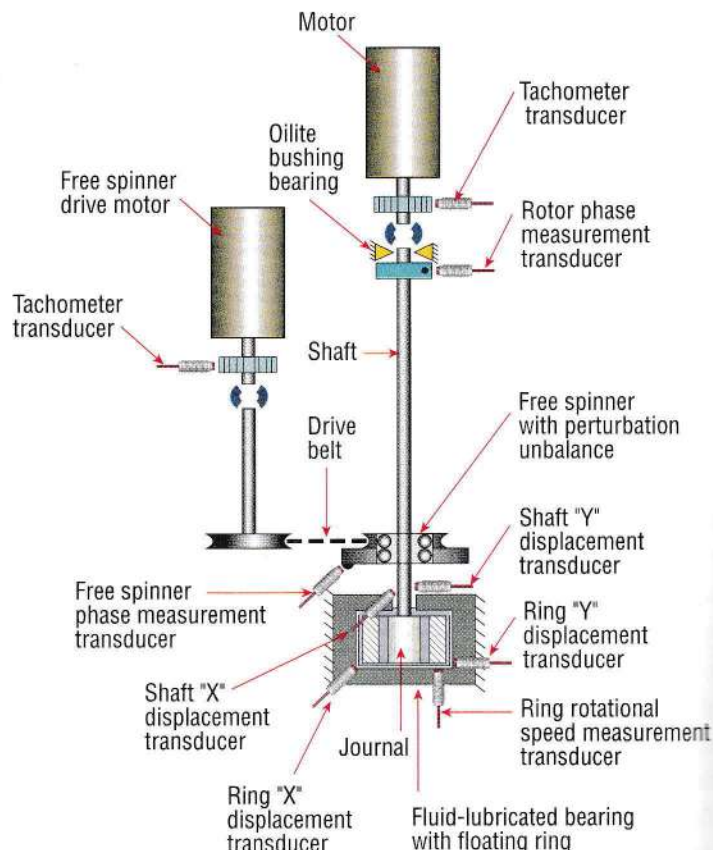


Figure 3. Experimental rig with the floating ring in a fluid-film bearing.

Quality Assurance at Bently Nevada



by Lorri Williams

Quality Assurance Manager
Bently Nevada Corporation
e-mail: lorri.williams@bently.com

It was not long ago that most manufacturing companies practiced "quality control" – a culture of always inspecting and checking to make sure everyone was doing his/her job. That culture has long since passed at Bently Nevada Corporation. In the 1970s and 1980s, Bently Nevada had a very large and task-oriented Quality Control organization. In the 1990s and continuing through today, the quality focus has changed from one of "control" to one of "assurance."

Bently Nevada shifted away from in-process inspection and expanded its quality focus to include design, manufacturability, and service. The changes integrated quality into each of the processes, utilized defined checkpoints, and made sure the products and services provided have the foundation to meet the stringent quality expectations of our customers (both internal and external). The focus and development of these changes has been shared by the Quality Assurance organization as well as the company's other organizations. As a result, quality has become a line function of the overall operating methodologies and strategies within the company.

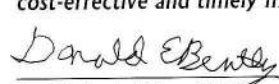
Our pursuit of ISO 9000 Certification in the early 1990s provided an opportune test of our recently changed quality system. Bently Nevada sought to certify each of our design and manufacturing operations. Through a total company effort, our Minden, Nevada, and Houston, Texas, sites in the U.S., and our Warrington, England, site, were all certified to ISO 9001 by 1993. In addition, our Bently Nevada Australia office was certified to ISO 9001 in 1995. We have continued to maintain the certifications in each of these locations.

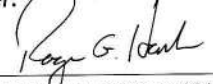
We have been successful in our quality efforts for many reasons, among them the unwavering support of top



QUALITY POLICY

Bently Nevada Corporation and its subsidiaries are committed to provide our customers with safe, defect-free products and services. Quality at Bently Nevada is a line function, in that it is each person's responsibility to produce quality work. It is management's responsibility to provide the required organization, direction, training and resources. We take pride in our ability to design, manufacture, install and service quality products in a cost-effective and timely manner.


 Donald E. Bently, Chairman & CEO


 Roger G. Marker, President & COO

management and the overall commitment of employees. The support starts at the top and is well documented by our Quality Policy, a policy that hangs in prominent locations throughout our facilities.

Quality Assurance is presently comprised of 21 employees, including engineers, technicians, and administrative personnel, who work together with Manufacturing, Design, Sales and Service. The typical Quality Engineer has a degree in engineering and between 3 and 20 years experience in a manufacturing/design environment. These engineers are active members in the American Society for Quality (ASQ), and the majority of them hold one or more certifications through ASQ (i.e., Certified Quality Engineer, Certified Reliability Engineer, or Certified Quality Auditor). It is the strength and depth of these key employees that helps us to design quality into our products and processes.

Quality is integrated into the design early in the process. Doing so strengthens the likelihood that the released product will meet and continue to meet the product requirements and specifications. To accomplish this, a team of skilled Quality Technicians works together with Engineering to test and evaluate new products before and after release to production. These technicians maintain an extensive Engineering lab,

The raw data and the described data processing is illustrated in Figure 6. The results of all perturbation tests, reduced to the parameters of air-induced Dynamic Stiffness (Equations 5 and 6), are presented in Table 2.

The most significant conclusion from the results in Table 2 is that aerodynamic forces applied to the impeller generate negative spring stiffness (damping D_{air} and stiffness K_{air} appear negative in all experiments). It is also important that this stiffness drops even more during the rotating stall. This result is also shown in Figure 7. Inconsistency in tangential force coefficient T suggests that it was too small in the experiment and was not determined with sufficient accuracy.

Case history

A 5-stage propylene compressor provides an example of the industrial centrifugal compressor behavior under the conditions of surge/rotating stall. The orbit (Figure 8) shows a typical forward precession with frequency 0.15X. This is supported by the lateral vibration, half spectrum cascade data (Figure 9). The gas pressure measured at the outlet of the third stage is presented in the timebase waveform for the steady state regime (Figure 10) and half spectrum cascade plot for the startup (Figure 11). The steady state data shows the combination of vane passing high frequency with low frequency pulsations. Using the half spectrum cascade plot, the latter are

identified as a surge component (3.5 Hz) and rotating stall component (0.15X). It is important to note that the half spectrum cascade of the thrust probe data (Figure 12) shows a strong activity at the surge frequency, while the rotating stall component is much smaller. The described behavior is a reflection of the surge, which propagates axially, while rotating stall is mainly active in the circumferential direction.

Instrumentation and diagnostics strategy

A typical machine train for a centrifugal compressor unit is shown in Figure 13. It includes an engine (in this case an aeroderivative gas turbine), gearbox, and the compressor itself. Different instrumentation configurations for vibration, as well as performance diagnostics and monitoring of the engine, are beyond the scope of this paper. For the centrifugal compressor, the instrumentation package normally includes a Keyphasor® transducer, two sets of proximity probes in XY configuration installed at the inboard and outboard bearing locations (see Compressor DE and Compressor NDE in Figure 13), two axial transducers, and performance monitoring

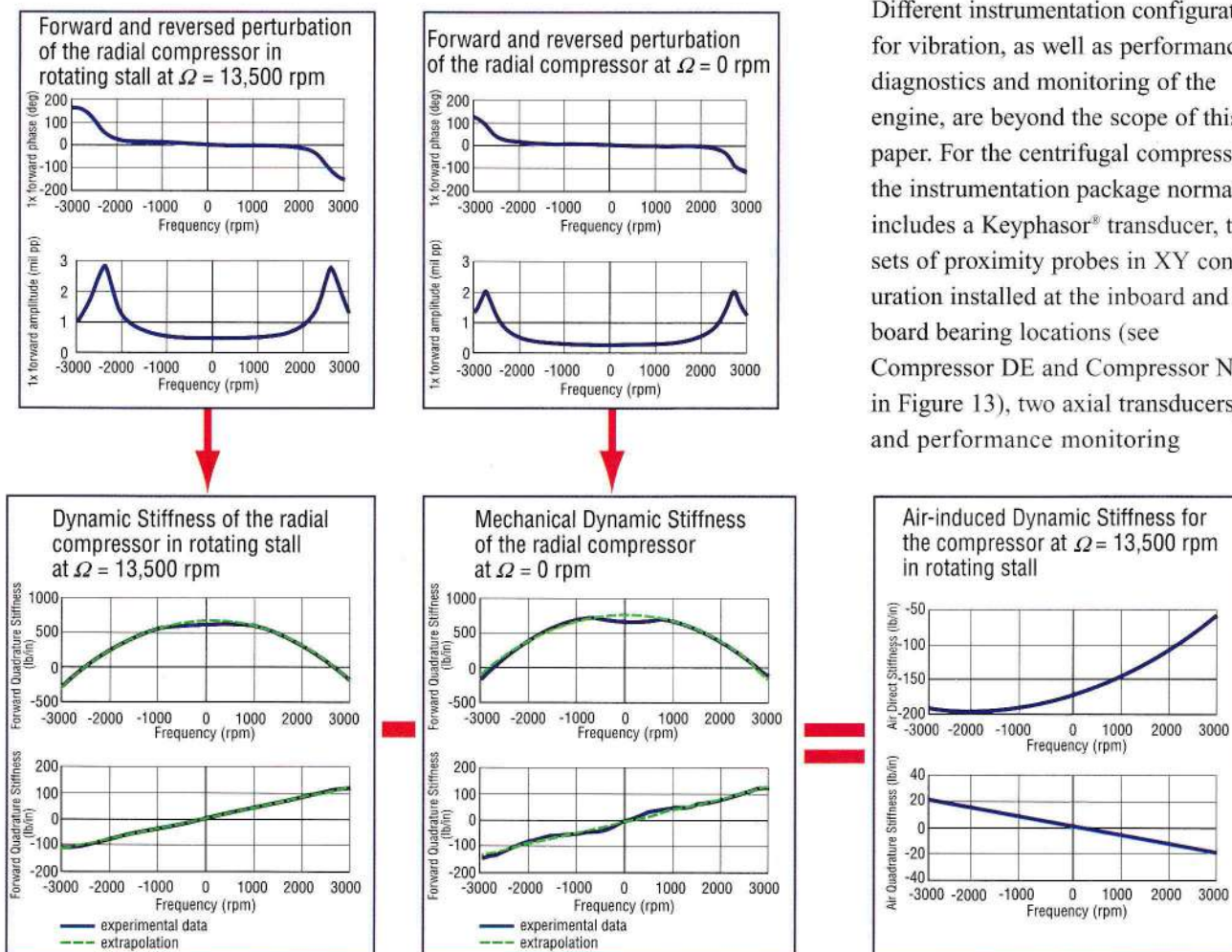


Figure 6. Identification of the air-induced Dynamic Stiffness component.

Dynamic Stiffness and the Advantages of Externally Pressurized Fluid-Film Bearings



by Donald E. Bently

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Most turbomachinery in use today employs oil-lubricated sleeve bearings. These bearings have evolved over the last century, but virtually all share the following characteristics:

- They are partially, rather than fully, lubricated.
- They are low-pressure; lubricant supply pressure is generally around 15 to 20 psi (103 to 138 kPa).
- They rely upon relative motion between static and rotating surfaces to generate the fluid film.

As we will see, bearings with the above three characteristics are generally thought to be more stable, and thus inherently better, than externally pressurized designs. Recent research, however, indicates that the opposite is true: modern bearings using externally pressurized designs are inherently more stable and offer many advantages over conventional bearing designs. This article will examine both conventional and externally pressurized fluid-film bearings.

Discussion of the differences in fluid-film bearing types is intimately linked to a discussion of Dynamic Stiffness. While numerous papers describing Dynamic Stiffness have been published by Bently Nevada and Bently Rotor Dynamics Research Corporation, they rely primarily on rigorous mathematical descriptions and experimental results. Since the mathematical and experimental foundation for the topics in

this article are already well established, this article addresses Dynamic Stiffness in simple terms, relying on simple physical descriptions rather than complex mathematics. [Editor's Note: For those unfamiliar with the quantitative details of Dynamic Stiffness calculations and derivations, the next issue of ORBIT will feature an extensive tutorial, complete with examples.]

Terminology

When referring to fluid-film bearing types and principles of operation, various authors use different terminology, which can sometimes result in confusion. Before proceeding, a review of terminology used in the literature, and a clarification of the terminology used and preferred by Bently Nevada, is helpful.

- **Bearing** – The stationary surface supporting a rotating shaft, in which the journal rotates.
- **Journal** – The rotating surface on the shaft that is designed to be in contact with the fluid film in the bearing clearance.
- **Fluid-Film Bearing** – A bearing in which the shaft is supported by a thin layer of lubricating fluid, and the rotating and stationary parts are not in direct contact with one another. Other bearing types, such as dry bearings and rolling element bearings, exist.
- **Rolling Element Bearing** – A bearing in which the shaft is supported by metal-to-metal contact between the rotating and stationary parts. This is done by using rolling elements that minimize friction. These bearings are usually lubricated to further reduce friction and to provide cooling.
- **Anti-Friction Bearing** – Any bearing in which moving parts are designed to touch one another, whether rolling element bearing, dry bearing, or other.
- **Dry Bearing** – A design in which the journal rotates directly on the bearing surface without any intervening fluid or rolling elements. Special low-friction materials



Beware of the Use of a Floating Ring in a Fluid-Film Bearing



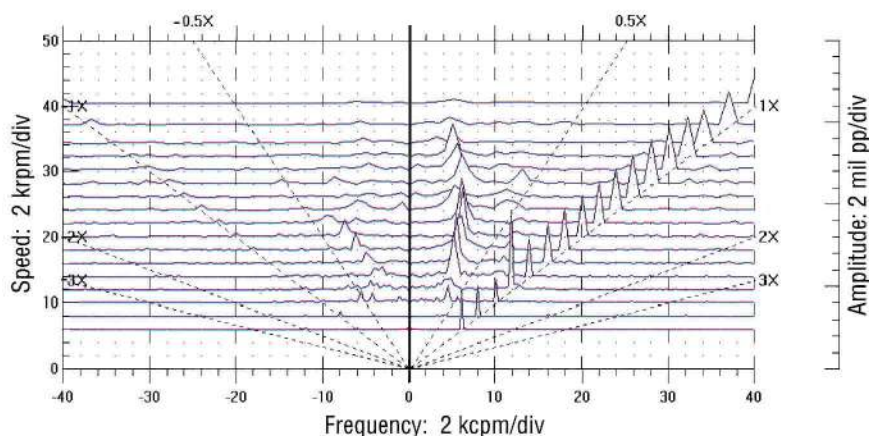
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Reverse vibrational components X to Y Rotation Forward vibrational components

Figure 1. Full spectrum cascade plot of a typical startup of a gas turbine with a fluid-film bearing with floating ring design.

Unloaded fluid-film bearings have the reputation as occasional sources of high vibration due to fluid-induced instability. When a bearing design becomes more complex, some additional types of instability may occur. In one type of fluid-film bearing (unimproved since its introduction 30 years ago), a floating ring separates the bearing from the journal. This brings up additional instabilities of the ring, in lateral and axial directions, plus provides unusual and difficult-to-predict motion of that ring. Not only do new instabilities appear, the old ones change their appearance in the presence of a floating ring, which leads to local lubricant starvation. Experimental results from the test rig, combined with data from the field, prove a floating ring in a fluid-film bearing to be useless and dangerous.

Fluid-film bearing with a floating ring

In this floating ring fluid-film bearing, the floating ring separates the bearing from the journal [1, 2]. The major objective for using a floating ring is a reduction of the lubricant speed gradient within the bearing. Particularly, it is assumed that the ring would rotate at a speed approximately half that of the journal. Additionally, a floating ring is supposed to "cure" such technical problems as misalignment.

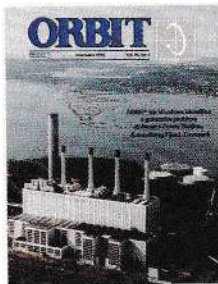
Meanwhile, machinery with fluid-film bearings with the floating ring design has been continuously troubled, including a catastrophic failure of a gas turbine [1]. As a result of the failure, impellers of the compressor, as well as the proximity probe for phase measurement, were damaged. The brass thrust washer was destroyed (melted)

and oil within washer passages was charred.

During startup of the gas turbine, an unusual phenomenon was observed (Figure 1). Fluid whirl appeared at a frequency less than $\frac{1}{2}X$. As the machine came up in speed, the whirl disappeared. At operational speed there was no whirl.

The plot also indicates the presence of low frequencies, which are typical for floating ring design of a fluid-film bearing. The disappearance of the fluid whirl is unusual. At this point it is important to mention that fluid whirl can exist only under the condition of sufficient lubrication. In this case, however, the floating ring impaired oil delivery. The whirl ceased due to the centrifuge effect that pumps lubricant out of the bearing through radial ports

same issue continues exploration of a new plot type pioneered by Bently Nevada in the article *Using Full Spectrum Plots – Part 1*. (Part 2 of this article continues in the June 1994 issue.)



1994

The June issue contains a nice review of the basics in *Rotating Machinery Measurements 101*, written by Don Bently and typical of his passion for teaching the fundamentals to our customers so that they are better able to apply our products and derive value from them.

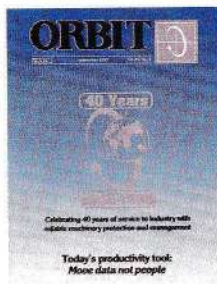
The September issue introduces our 3300/53 Overspeed Protection System, signifying Bently Nevada's entry to the overspeed detection marketplace on a worldwide basis.

1995

In the March issue, Don Bently makes his first published *Statement on Magnetic Bearings*, foreshadowing his later introduction of an actively controlled fluid bearing in 1999. In the same issue, an online version of our Engineer Assist™ product is announced, furthering our long-term strategy to bring these automated Decision SupportSM capabilities to the same number of plants as are already using our online data acquisition products.

In June, Bently Nevada announces availability of company and product information on the Internet, and *Forward and Reverse Orbits* are explored in an article by Don Bently.

The September issue introduces our new 3500 Series Machinery Protection System. It also contains an important message from our President, Roger Harker, on *Move Data, Not People®* about the benefits of using communications technology to remotely collect and diagnose machinery data. Rather than moving people and equipment to the source of the data, the data moves to the people – faster and at reduced cost. It's a vision that will touch almost every Bently Nevada product, as well as our approach to service offerings, over the next five years. Richard Thomas contributes an important Back-to-Basics article, *The Importance of Transient Data Analysis*.

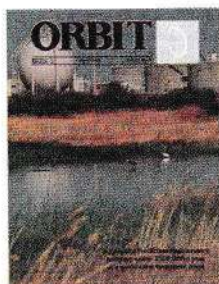


1996

A series of articles on fluid-induced phenomena starts off the March issue: *The Description of Fluid-Induced Whirl*; *Synchronous Amplification Factor, Q_s* ; and *Fluid-Induced Instabilities of Rotors*.

In June, the majority of the issue is devoted to protecting and managing reciprocating compressors. We also introduce Data Manager® 2000 for Windows NT®, the 5th-generation product of its kind, building on a history of over 1800 supplied online data acquisition systems. Finally, we announce our own website, www.bently.com, signifying our commitment to this emerging medium for “instant” communications and information.

In September, we address the emerging dialog in industry regarding “online asset management” with an excellent article by Marco Alcalde outlining *Asset Management Strategies*, and continuing in the December issue with an article I wrote on the subject titled *Asset Management – The New Challenge*.



1982

Our Winter issue continues the topic of using all available data to establish root cause, rather than just treating machinery malfunction symptoms, with the article *Cure the Machine of Pneumonia ... and It Dies of Diphtheria*. *Bently's Corner* addresses a message you'll hear often from us: vibration is simply a ratio of forces divided by stiffnesses. Therefore, understand why vibration changes, rather than assuming that "less vibration is good."

The July issue continues on the topic of cracked shaft in *Bently's Corner*, while the November issue features recommendations for *Monitoring Rolling Element Bearings*.

1983

The March issue introduces our "T-diagram," showing how our instrumentation is aimed at providing information to both machinery specialists and operators. The *Bently's Corner* article is called *Meaningful Machinery Information* and reflects our commitment to not just protecting machinery, but providing the information necessary to manage machinery as well.

The June issue provides an informative article called *Shaft Versus Housing Measurements* that offers guidance on choosing the correct transducer for specific applications. It underscores our commitment to provide all necessary machinery transducers (not just proximity probes) while teaching our customers how and when to apply them. The topic is continued in the July 1984 issue with *Housing Measurement Transducers*.

The October issue marks the beginning of published articles on machinery rub phenomena with a *Bently's Corner* devoted to the topic. As mentioned above, less vibration is not necessarily always favorable. The article *Low Vibration Levels Do Not Always Indicate a Healthy Machine* explores this topic further with a case history.

1984 / 1985

Perturbation Testing: Determining the Safety Margin of a New Machine leads off the March 1984 issue and addresses this important, but underused, technique for accurately modeling machinery in the field (not just the test stand). It's a topic we are still emphasizing today.

In the July 1984 issue, the article *What Happened to the Damping?* begins the first of numerous ORBIT articles exploring Dynamic Stiffness (both direct and quadrature terms) and introduces the term λ (lambda) for the very first time in ORBIT — a topic about which Don Bently and Agnes Muszynska will later publish extensively.

1986

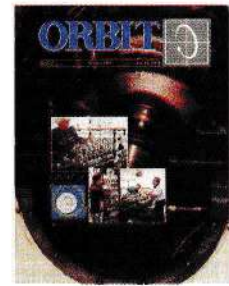
The January issue provides extensive resources on understanding and detecting shaft cracks with the articles *Vibration Analysis Techniques for Detecting and Diagnosing Shaft Cracks* and *Anatomy of a Shaft Crack*. For those who have experienced this significant problem, you know how important it is to detect the condition as early as possible. Bently Nevada research has resulted in proven methodologies and products aimed specifically at early shaft crack detection.

The June issue explores *Rotor Mode Shape Analysis for Assessing Machinery Condition*, a topic as relevant today as when first published, and a recently



1991

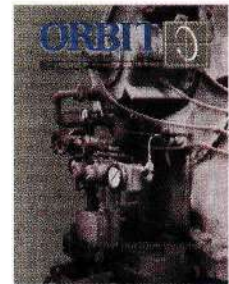
Bently Nevada enters the torque monitoring market in the February issue with the release of our TorXimitor® product for online measurement of torque and power. This will eventually lead to our entry into the performance monitoring market in 1997 with Performance Manager 2000. During this time, more and more case histories submitted to ORBIT rely upon data from our online systems such as Transient Data Manager®, indicating a shift away from intermittent, portable methods (such as ADRE®) as the preferred way to collect data on critical machinery.



1992

In the September issue, we introduce Engineer Assist™ software, the first product of its kind for moving beyond just online data acquisition and plot displays, and instead harnessing the power of the computer to reduce the data into meaningful information. Like Trendmaster® 2000, this product was probably a decade ahead of its time, with significant interest in such automation only recently beginning to materialize as plants reassess how such systems can help them achieve competitive advantage by making better use of limited human resources.

The 2201 Monitoring System is introduced in the September issue as well, signifying Bently Nevada's ability to tightly integrate our solutions with control and automation equipment, such as Allen-Bradley's PLC-5® family of programmable controllers. More information on the *Performance of REBAM® During Ball Bearing Failures – Part 1* is presented, further justifying the use of proximity probes as a viable approach to these types of bearings. (*Part 2* of this article appears in the December 1992 issue.)



Finally, an innovative measurement parameter termed "Not 1X" by Bently Nevada, is introduced in the September issue. It would later find its way into virtually all our software products, and hardware such as 3500, ADRE®, and Snapshot™ for Windows® CE. You can read about it in the Probe Tip titled *The Importance of Not 1X Vibration*.

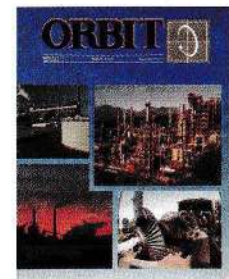
1993

In the March issue, Don Bently and Charlie Hatch offer some *Precautions on Polar-Plot Balancing*, and Agnes Muszynska explores *Thermal Rub Effect in Rotating Machines* with an excellent mathematical analysis of this common machinery problem.

In June, Bently Nevada introduces the Plus and Minus Spectrum (later to be called Full Spectrum), a significant enhancement to frequency domain analysis of vibration. Al Campbell provides the article *Static and Dynamic Alignment of Turbomachinery*, one of numerous articles by Bently Nevada on this subject, and one which emphasizes our extensive knowledge and capabilities to provide machinery alignment.

The September issue sees Don Bently offering *Historical and Future Views of Rotor Dynamics* and alluding to the use of root locus techniques for rotor system analysis.

The December issue provides a more extensive treatment of root locus in the article *Root Locus and the Analysis of Rotor Stability Problems*, showing why this technique is so powerful and necessary for any modern machinery engineer. The



Rotative speed	Parameters	No rotating stall	With rotating stall	Stiffness change, %, relative to normal conditions
11,500 rpm	K_{air} lb/in	-52.16	-63.2	-21
	D_{air} lb•s/in	-3.4×10^{-3}	-3.0×10^{-3}	
	M_{air1} lb•s ² /in	-2.7×10^{-6}	-3.0×10^{-6}	
	M_{air2} lb•s ² /in	-9.75×10^{-6}	-9.7×10^{-6}	
	T lb/in	2.83	3.6	
13,500 rpm	K_{air} lb/in	-134.6	-186.4	-38
	D_{air} lb•s/in	-7.0×10^{-3}	-8.3×10^{-3}	
	M_{air1} lb•s ² /in	-4.3×10^{-6}	-7.7×10^{-6}	
	M_{air2} lb•s ² /in	-10.0×10^{-6}	-7.7×10^{-6}	
	T lb/in	-0.52	-0.07	
15,000 rpm	K_{air} lb/in	-66.7	-134.1	-101
	D_{air} lb•s/in	-5.7×10^{-3}	-5.7×10^{-3}	
	M_{air1} lb•s ² /in	-4.3×10^{-6}	-4.3×10^{-6}	
	M_{air2} lb•s ² /in	-10.0×10^{-6}	-10.0×10^{-6}	
	T lb/in	-0.52	-0.52	
17,000 rpm	K_{air} lb/in	-63.2	-130.8	-107
	D_{air} lb•s/in	-5.2×10^{-3}	-6.4×10^{-3}	
	M_{air1} lb•s ² /in	-1.5×10^{-6}	-5.0×10^{-6}	
	M_{air2} lb•s ² /in	-8.7×10^{-6}	-8.3×10^{-6}	
	T lb/in	3.6	2.7	

Table 2. Parameters of the aerodynamically-induced Dynamic Stiffness.

instrumentation. As a minimum, the compressor performance monitoring instrumentation is supposed to include inlet and discharge pressures and temperatures, and discharge flow measurements. Using the described instrumentation, the rotating stall or surge can be identified based on the following information:

- **Radial vibration**

Steady state data: direct orbits, full spectrum

Indication: subsynchronous forward component

Possible causes: **rotating stall**, fluid-induced instability, and surge

Transient data: direct orbits, full spectrum cascade

Indication: subsynchronous forward component with frequency proportional to the rotative speed (0.1 to 0.8X)

Possible causes: **rotating stall**, fluid-induced instability

- **Axial vibration**

Steady state data: spectrum

Indication: subsynchronous component

Possible causes: **rotating stall** (possible weak coupling), and surge

Transient data: spectrum cascade

Indication: subsynchronous component with low constant frequency

Possible cause: **surge**

- **Pressure, temperature, flow**

Trend data (dc signal)

Indication: pressure, head, and flow decrease, temperature increase

Possible causes: **surge**, less likely rotating stall

Note: if pressure head and flow indicate a position on the performance curve close to the surge point, it could support rotating stall or surge versus fluid-induced instability

Dynamic data

Indication: vane passing frequency amplitude modulation of pressure, with constant flow

Possible cause: **rotating stall**

Indication: pressure vane passing frequency amplitude modulation with correlated flow oscillations

Possible cause: **surge**

As indicated above, a strong conclusion can only be reached by the correlation of vibration data with performance information. It is important to note that rotating stall can originate in a particular stage of a multistage compressor. To localize the rotating stall source, additional pressure transducers may be required.

Conclusions

Although, the rotating stall aerodynamic effects have been investigated extensively [16-23], the rotor dynamic implications of this phenomenon have received much less attention [24, 27-29]. A direct quotation from the paper [26] by Colding-Jorgensen serves as an example of insufficiency of rotor dynamic data:

"In spite of all reservations due to difficulties and lack of data, it is the author's opinion that the impeller-casing effect is mainly responsible for a negative direct stiffness effect."

Vibrational Diagnostics of Rotating Stall in Centrifugal Compressors



by Donald E. Bently

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This paper discusses the rotating stall in centrifugal compressors from vibrational diagnostics and rotor dynamics standpoints. A diagnostic strategy is suggested from the overview of the case history and aerodynamic research. The literature survey shows insufficient understanding of rotor dynamic implications of the rotating stall. Most of the literature on the subject addresses only aerodynamic aspects of the problem, and only a few papers treat a compressor as a coupled mechanical-aerodynamic system. The latter approach is exercised in this paper. The results of the rotor/fluid system Dynamic Stiffness identification are presented for the experimental centrifugal compressor, in rotating stall and in normal operating conditions. It is shown that the aerodynamically-induced radial (direct spring) stiffness coefficients are

negative in normal operating conditions. The negative spring effect in the case of rotating stall is even more profound. The conclusion is that during rotating stall the rotor dynamic stability margin is reduced.

Introduction

The interest in radial compressor fluid-induced instabilities started in the mid-seventies. Since then, numerous case histories of high pressure, centrifugal compressor instabilities have been published [1-17]. Most of the publications report two types of rotor vibrational behavior: 1) High eccentricity, rotor first natural frequency re-excitation; and 2) Subsynchronous forward precession with rotative-speed-dependent frequency. The former is usually referred to as whip-type behavior and is normally associated with balance pistons, fluid-film bearings, and

labyrinth seals. The latter is called whirl-type behavior [27-28] and can be associated either with fluid-film bearings/seals or with rotating stall. Rotating stall usually results in an appearance of a low subsynchronous frequency component in the rotor vibration spectrum (frequency ratios are typically between 8 and 40% but can be as high as 80% of rotative speed, [20]). The emphasis on rotor behavior does not allow for details of particular flow patterns but treats the fluid (gas) "in average," which falls in the scale of rotor motion.

Experimental compressor

An experimental rig (Figure 1 and Table 1) was constructed to evaluate the rotor/fluid system Dynamic Stiffness during rotating stall and normal operating conditions. The rig consisted of a single stage, centrifugal

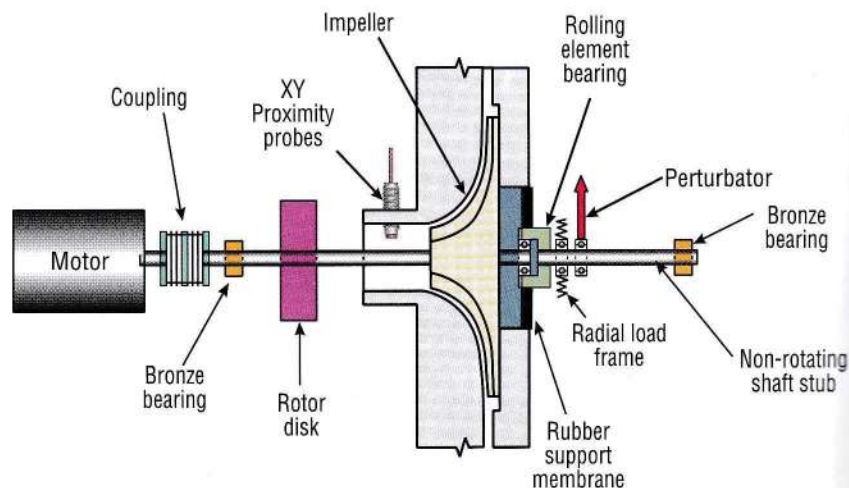


Figure 1. Experimental compressor.

The December issue contains an article that outlines *Best Practices for Using Monitoring System Outputs*. The article has been available on our website for quite some time because of requests for reprints and its attention to a topic that had not been previously addressed by Bently Nevada. It represents a continuing evolution of how our products are integrated to other control, protective, and information automation systems.

1997

The March issue kicks off with some informative articles on the use (or should we say misuse) of vibration "transmitters." The first deals with the economic considerations and is titled *Vibration "Transmitters" Cost Less Than Monitors ... Fact or Fiction?* The second deals with the practical application issues and is titled *The Limitations of Protecting and Managing Machinery Using Vibration "Transmitters."* These articles signify our continued emphasis on choosing instrumentation that permits both machinery protection and machinery management.

The September issue introduces two important new products: MachineLibrary™ and Machine Condition Manager™ 2000. The first is a multimedia reference tool that embodies over four decades of accumulated knowledge on rotating machinery behavior and measurements. The latter represents the evolution of our Engineer Assist™ product to a Microsoft Windows®-based product that is tightly integrated with our Data Manager® 2000 software. Again, it underscores our commitment to making Decision Support™, rather than just plots and data, the new standard for online software system capabilities. Finally, the September issue marks Bently Nevada's entry to the thermodynamic performance monitoring market with the introduction of Performance Manager 2000, again demonstrating our evolution to an "all your machinery" company capable of making more measurements on machines and also addressing more machine types. In fact, our 2nd-generation performance monitoring product, Bently PERFORMANCE™, is just about to be released.

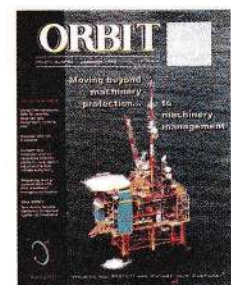
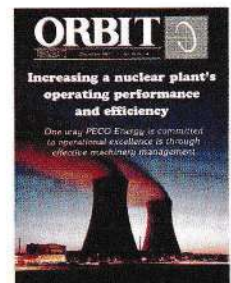
1998

Further refinements of Dynamic Stiffness concepts are presented in *Dynamic Stiffness in Whirl and Whip*, appearing in the March issue. In June, we provide an informative *Interview* with our owner, founder, Chairman, and Chief Executive Officer, Don Bently, followed by a similar *Interview* in the December issue with our President and Chief Operating Officer, Roger Harker. Together, these interviews provide excellent insight into where we are today, where we are going, and how we have evolved as a company over the past forty-five years.

1999

The First Quarter issue provides an important glimpse into Bently Nevada's principle-centered culture with Don Bently's article on our *Core Values*.

The Second/Third Quarters issue introduces the new Bently ServoFluid™ Control Bearing, the most innovative product in the company since the introduction of the proximity probe. A series of articles addressing this product are included. The first is titled *Active-Control Fluid Bearings for a New Generation of Machines*. The second is titled *Magnetic Bearings ... Do Their Problems Outweigh Their Benefits?* The important topic of *Root Locus Analysis* is re-visited, and an



System 1™ and Snapshot™ for Windows® CE Setting the standard ...



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System 1™, Bently Nevada's exciting new machinery management platform, will soon be released. Over the past year, Bently Nevada has been talking about the development of this open and integrated platform with its single display environment, open database, and multiple means of data acquisition. This initial release offers portable data collection as the first method of data acquisition that we are supporting. Continuous online data acquisition support will follow later in 2000, along with Decision SupportSM and performance monitoring.

Incorporating the features of Bently Nevada's many stand-alone software applications for machinery management, System 1™ integrates these separate applications and adds greatly expanded functionality, enabling machinery management professionals to manage all their machinery assets with a single system. As a platform, System 1™

will integrate a host of condition monitoring devices and technologies into a single system with a unified display environment and common database structure, resulting in reduced training requirements for users and lower implementation and support costs.

As well, System 1™ will provide a variety of industry-standard interfaces including OLE for Process Control (OPC), enabling external integration with a multitude of control and automation applications. This external integration provides the ability to correlate data and information both within System 1™ and within other complementary control and automation applications, including human interfaces, data historians, maintenance management systems, and others.

System 1™ has been developed for the Microsoft® Windows NT® operating system, taking advantage of the substantial installed base, the intuitive interface, and the many features of this industry-standard operating system. Just one operating system to lower system-sustaining costs!

This initial release provides support for our new Snapshot™ for Windows® CE portable data collector and analyzer. However, System 1™ will rapidly evolve to incorporate both online and offline modes of data acquisition, with

support for many of our communications

processors such as TDXnet™ and DDI (Dynamic Data Interface), as well as our Trendmaster® 2000 system hardware. Other machinery management tools, like balancing software, and technologies, such as oil analysis, are also planned for inclusion in System 1™.





On the Lighter Side



by Donald E. Bently

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Several months ago, I was asked what I'd like to say in this 20th anniversary issue of the ORBIT. As I pondered the question, I provided some direction on several of the articles I wanted to see in this issue, such as the articles on *Rotating Stall* on page 32 and *Dynamic Stiffness* on page 18. I also helped identify some of the past articles we wanted to highlight in this issue. However, I've never been one to dwell on the past too much. Mostly, I'm interested in the future, and I didn't want this issue of the magazine to focus on our past to the extent that it eclipsed the exciting things happening today and in the future. Those who know me know that I live in the present, not the past. Case in point: I'm just as passionate today about our new ServoFluid™ Control Bearing as I was in the early days of this company with our proximity probe technology. I'm anticipating the same effect on industry with this new technology as we have achieved with the proximity probe over the last 45 years.

Returning to the question asked in the first sentence, I actually have a few

rather lighthearted observations I have wanted to make for quite a long time. Taking a page or two in this anniversary issue seemed a fitting place to do so. I hope they'll bring a smile or two, and I hope they'll provide some food for thought.

Newton's Apple ...

Newton was a smart guy. Imagine being known as the co-inventor of Calculus as just one of your many achievements. In anyone's estimation, he was a genius of the first degree. I often hear his 3rd Law of Motion paraphrased as, "For every action, there is an equal and opposite reaction." Now maybe it's just the engineer in me, but I know that Newton stated his law in terms of forces...not "actions" and "reactions," and I prefer to speak of this law as Newton intended: in terms of forces and moments. They are vector quantities, and the physics of motion are vector quantities, as any billiards player can attest (the ball must have the correct magnitude *and* direction).

Speaking of vector quantities, that reminds me of another topic I've been meaning to write about in this publication for years ...

Real and Imaginary ...

We engineers use vector quantities in one form or another in hundreds of ways. For two-dimensional vector quantities, it is typical to use complex numbers to perform vector computations.

Mathematicians talk about complex numbers and vector quantities by referring to their "real" and "imaginary" components. In fact, that is the nomenclature used in almost every textbook and classroom in the world. I suppose I'm a maverick, because for years I've avoided the use of these terms, preferring instead to use the terms "direct" and "quadrature" when talking about vector components and complex quantities. Why? I suppose I object to the idea of any physical quantity, be it vibration, electric current, Dynamic Stiffness, or others, as being somehow "imaginary" just because one component is 90 degrees out of phase with the other. That's why I use the "direct" and "quadrature" terminology; it conveys that one component is every bit as "real" as the other. Compelling evidence of this is as near as your closest electrical outlet. Build a suitable circuit with inductors and capacitors, plug it in, and you'll get both "real" and "imaginary" current components alright; but, please don't touch the conductors. You'll find out quickly that there is nothing "imaginary" about this electricity, regardless of its phase relationship to the outlet's voltage!

You'll notice this "direct" and "quadrature" nomenclature throughout our publications and technical papers, and you'll note a distinct absence of the terms "real" and "imaginary." Now you know why.

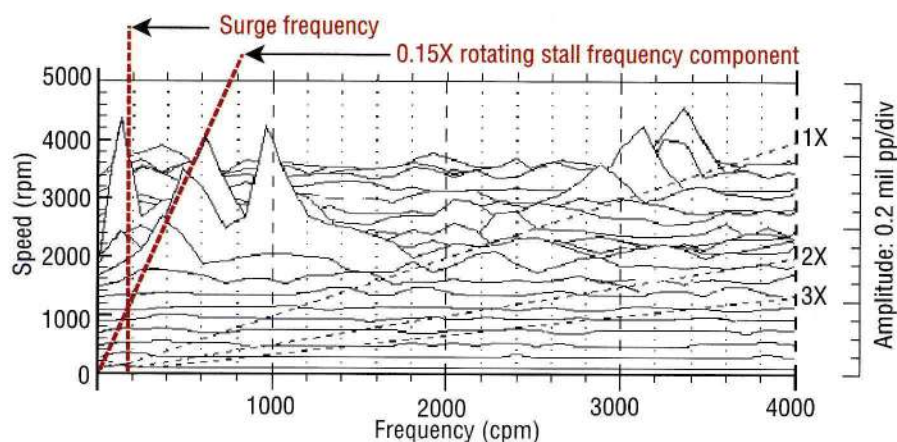


Figure 11. Half spectrum cascade plot of gas pressure oscillations at the outlet of the third stage of 5-stage propylene compressor during startup.

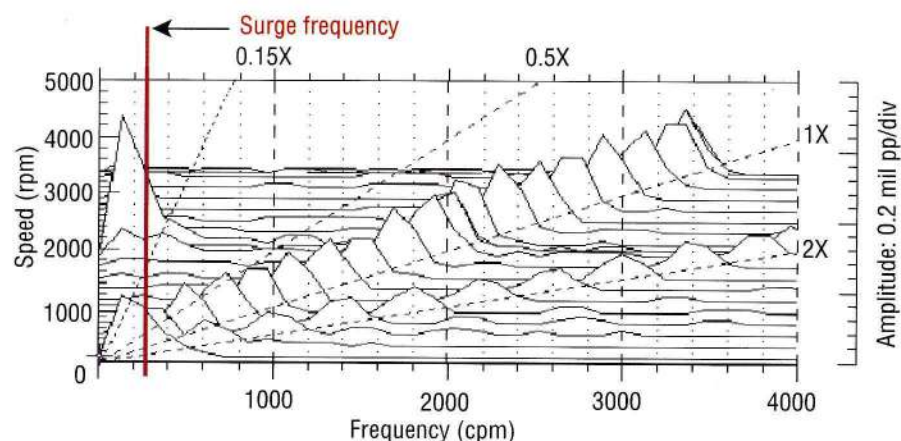


Figure 12. Half spectrum cascade plot of the rotor axial vibrations of the 5-stage propylene compressor during startup, showing surge component.

have lower natural frequencies than follows from purely mechanical calculations.

- The aerodynamically-induced Direct Stiffness during rotating stall drops dramatically and ultimately can make a compressor unstable.
- Altogether, aerodynamic forces may have a significant destabilizing effect on the compressor. In the case of a flexible rotor and/or soft bearings, they are able to create a rotor dynamic instability of whirl type. $\odot$

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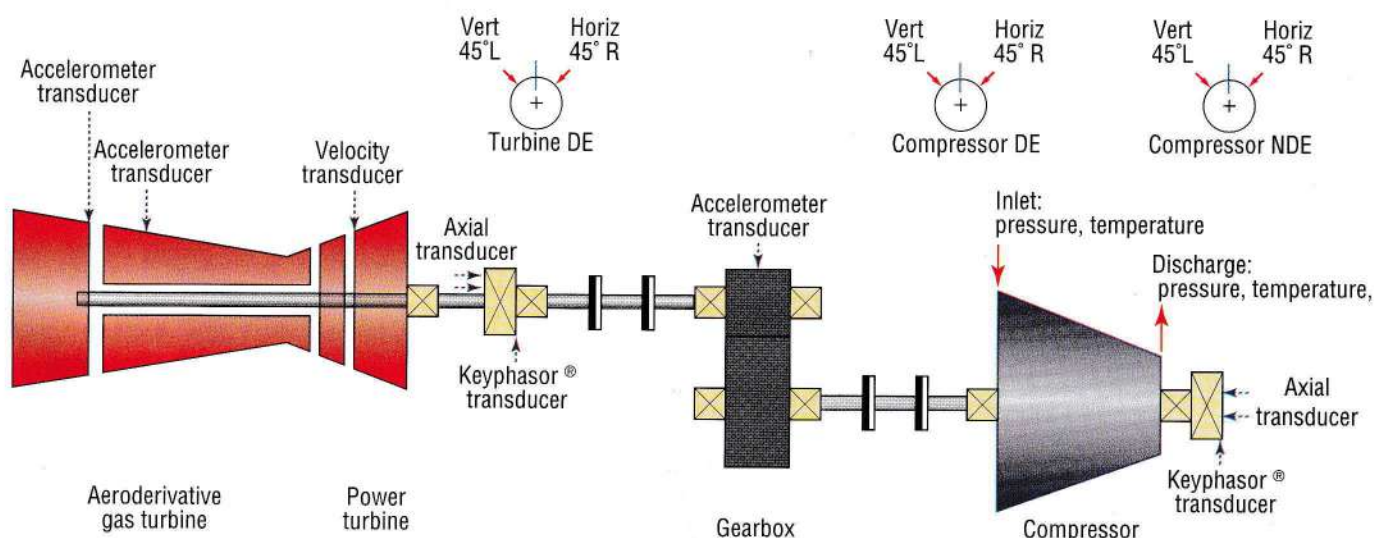
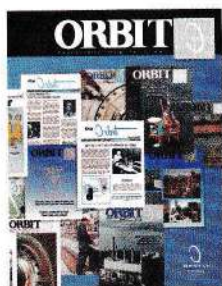


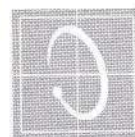
Figure 13. Typical compressor train diagram.



Cover: 20 Years of ORBIT
See From the Desk of
on page 8.

ORBIT

A Technical Publication of Bently Nevada Corporation



First Quarter 2000

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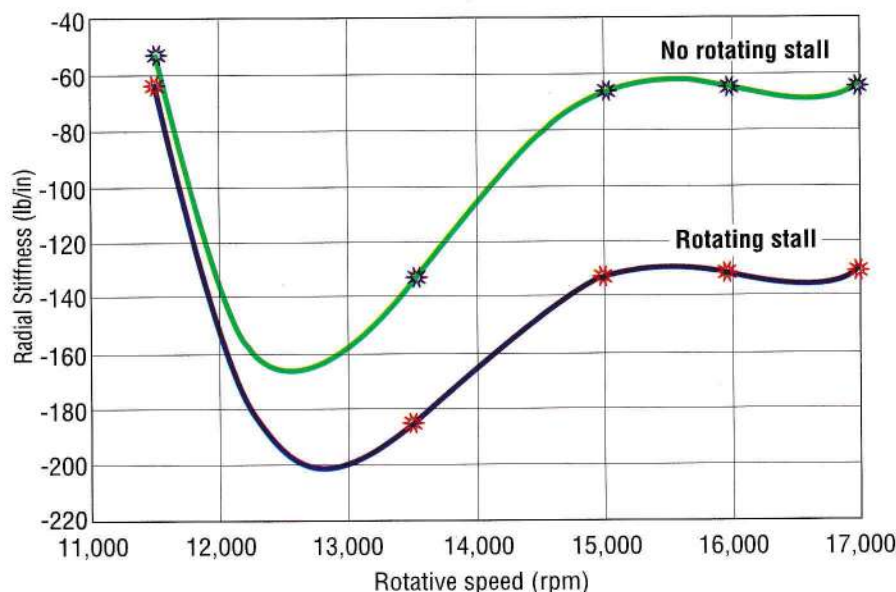


Figure 7. Aerodynamically-induced radial stiffness of experimental compressor.

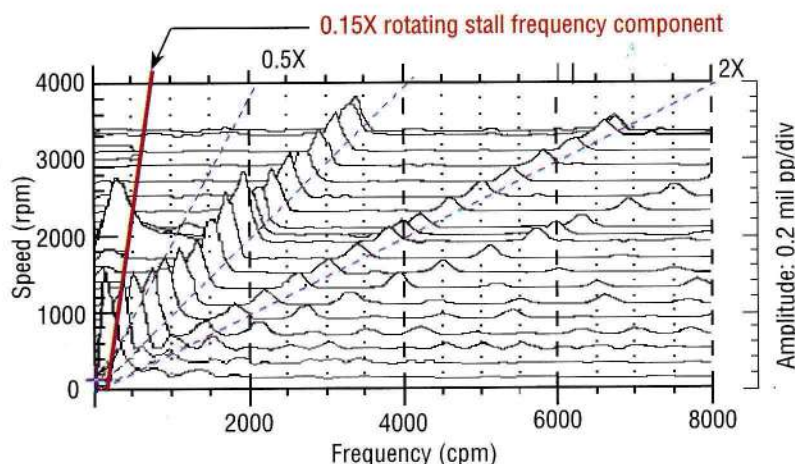


Figure 9. Half spectrum cascade plot of the vertical inboard lateral vibrations of the rotor at the outboard bearing of the 5-stage propylene compressor during startup.

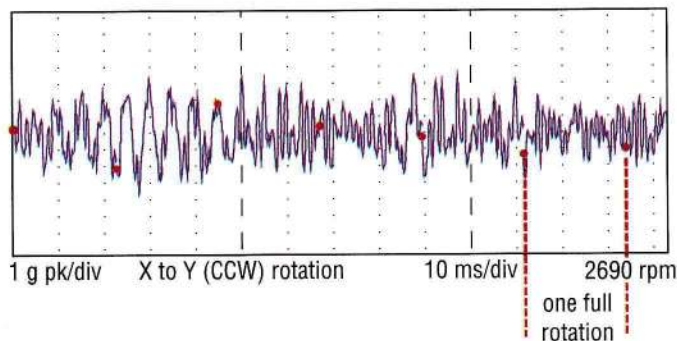


Figure 10. Gas pressure oscillations measured at the outlet of the third stage of 5-stage propylene compressor in steady state regime. Note high vane passing frequency modulated by the low frequency of rotating stall/surge.

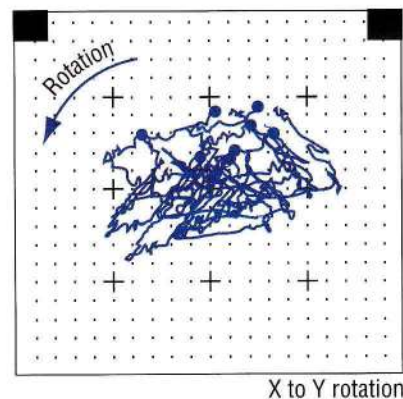


Figure 8. Typical orbit of a 5-stage propylene centrifugal compressor rotor in rotating stall conditions.

This paper provides some practical recommendations on the rotating stall diagnostics as well as some insight into rotor dynamic implications of the phenomenon. In short, the conclusions can be formulated as follows:

- The rotating stall conditions manifest in the rotor vibration signature as rotor forward precession with subsynchronous frequency, which tracks the rotor speed. This behavior is referred to as a rotating stall whirl.
- From the diagnostic standpoint, rotating stall differs from the other instabilities of the whirl type in its strong dependence on the compressor operating conditions; it should disappear with increased flow. It differs from surge by its frequency proportionality to the rotative speed as well as by the predominantly axial character of surge-caused vibrations.
- The parameter clearly identified by nonsynchronous perturbation tests is aerodynamically-induced direct spring stiffness. In all conditions it was negative. From the practical standpoint, it means that even in normal operating conditions, centrifugal compressors

Looking Back ... on the Best of ORBIT



by Steve Sabin

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Looking back over twenty years of ORBIT magazines, I'm reminded of how we have evolved as a company, and how the messages emphasized in ORBIT convey this evolution. I'd like to reflect a bit on those twenty years and share some of the more important messages we've brought to you during that time by highlighting key articles. To keep the size of such an article manageable, we've chosen to focus primarily on those articles that represent our evolving knowledge of machinery behavior and best practice usage of our products.

Consistent with our belief that the past is important, but the present and future more important, we also will not take up space re-publishing the articles mentioned here. Instead, we invite you to visit our website at www.bently.com where we've conveniently provided online versions of these past articles.

1980

The premier edition of ORBIT is launched in the summer. The lead article? *Bently Nevada Introduces ADRE*. The release of this computerized data reduction system reflects the increasing emphasis of Bently Nevada on not just protecting machinery, but on understanding its condition so the machinery can be managed. As the years progress, Bently Nevada will evolve to more sophisticated versions of ADRE® and eventually our online systems that collect and reduce data continually. In the same issue, *Don Bently's Corner* emphasizes his interest in shaft crack phenomena, a topic on which he will later publish important discoveries and inventions.



1981

Our Spring/Summer issue features two articles of note. The first, *No Singular Piece of Data Can Reveal Problem*, reflects a simple but important truth: machinery diagnostics can rarely rely on a single plot or data type for a complete understanding of problems. Whether a half or full spectrum plot, orbit presentation, trend, or polar plot, always look for multiple indicators from various sources to confirm problems.

The other, *Breakthroughs Made in Observing Cracked Shafts*, is the first of numerous articles appearing in ORBIT that deal with Don Bently's discoveries regarding the behavior and detection of shaft cracks. It is typical of Don's approach to topics. Just a few months earlier, he had expressed an interest to our readers about these phenomena, and at this point he was already publishing the first of many articles regarding what he was learning.

In the fall, an article titled *A Formula for Verifying Cracked Shafts* appears, adding to the world's knowledge on this topic.

